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BALL VERSUS TAPERED ROLLER BEARINGS.

BY LIEUTENANT JAMES E. HAMILTON, U. S. N., MEMBER.

There is a large amount of literature on the subject of ball and roller bearings but nothing can be found wherein they are treated as competitive. Even in the catalogues of manufacturers, the discussion which is given is applicable to "anti-friction" bearings, which as a generic term covers ball and roller bearings of all types.

In this paper, an attempt is made to compare two types of "anti-friction" bearings for the purpose of determining if they are competitive or supplementary; that is to answer, from an engineering standpoint, the questions: can either ball or tapered roller bearings be used in each application; or are there two distinct fields, one where ball and the other where roller bearings should alone be used?

The points on each of which a comparison seems desirable and which are therefore covered in this paper are: dimensions, weight, radial load capacity, thrust load capacity, and speed. Other

points, which are not covered in great detail but which are mentioned, are lubrication and efficiency. Naval applications in particular were kept in mind throughout.

All of the data given in the tables are taken from the catalogues of the manufacturers and hence such comparisons should disclose the relative merits of the two types, since each is undoubtedly presented in its catalogue in the most favorable light possible.

The Navy Department Specification (42B5), covering ball bearings, lists five different types of radial bearings, each with three series of different weights. The General Specifications for Machinery of the Bureau of Engineering restrict the use of ball bearings for Naval machinery under its cognizance to two of these types, one single and the other double ball, and to the medium weight series of each type. Because of this restriction, these two types (one weight of each) of ball bearings were selected for comparative purposes, the dimensions being shown in Tables I and III, "single" and "double" ball respectively, together with other catalogue data.

The Navy Department not yet having published specifications for roller bearings, there were not available similar "Naval" roller bearings which could be used for comparison with the Bureau of Engineering "standard" ball bearings selected for discussion in this paper. Catalogues of various manufacturers were, however, available; and from the catalogue of a manufacturer of outstanding reputation and recognized engineering ability, two series of tapered roller bearings were selected, these two series being shown in Tables II and IV, "small" and "large" tapered roller bearings respectively. The series called "small" was selected for comparison with "single" ball bearings; and a somewhat heavier series, the one called "large," was selected for comparison with "double" ball bearings. The two roller series were so selected as to obtain bearings with dimensions as close to those of the ball bearing series as the catalogue offered.

The roller bearings of the two series are of the same general design and differ only in weight. Obviously, there is no such distinction between them as the "single" and "double" rows found in the ball bearings. A comparison of three series, one "single" ball, one "double" ball, and one of tapered rollers might have covered the subject but it was considered more desirable to make

the comparisons of two pairs of series: "single" ball versus "small" roller, and "double" ball versus "large" rollers respectively.

DIMENSIONS AND WEIGHT. FIGURES 1, 2, 3 AND 4.*

Although an attempt was made to select two series of tapered roller bearings of the exact dimensions of the "standard" ball bearings, a perfect agreement was found impossible. The outside diameters and widths plotted were taken directly from manufacturers' catalogues. For ball bearings, these dimensions agree exactly with those of the standards listed in the Navy Department Specification.

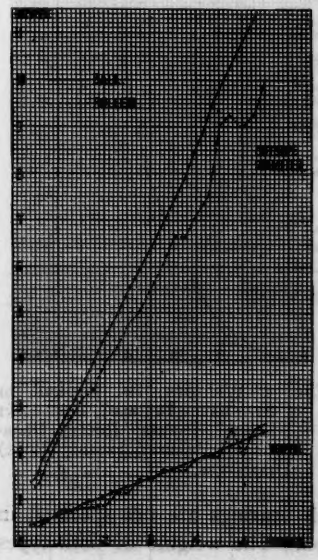

FIG. 1—WIDTHS AND OUTSIDE DIAMETERS OF "STANDARD SINGLE" BALL AND "SMALL" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

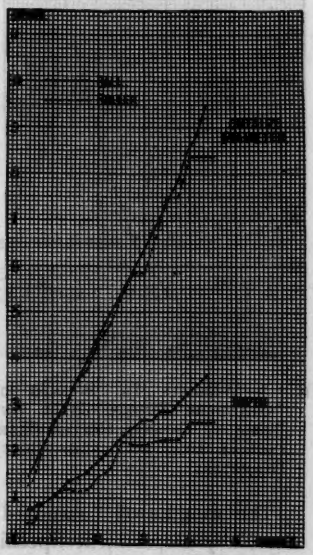

FIG. 2—WIDTHS AND OUTSIDE DIAMETERS OF "STANDARD DOUBLE" BALL AND "LARGE" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

* (NOTE: All curves in this paper except those of Figure 11 show the characteristic being compared plotted against inside bearing (bore) diameter or journal size because this dimension is the one which is used in selection of bearings for an application and probably offers the best norm upon which to base comparisons. On all Figures data for ball bearings are plotted in solid lines and for roller bearings are plotted in broken lines.)

The dimensions of the standards selected are tabulated in columns 1, 2 and 3 of Tables I, II, III and IV. The data are taken directly from catalogues. Outside diameter and width are both plotted against bore diameter in Figures 1 and 2. In both Figures the width is the lower of the two curves.

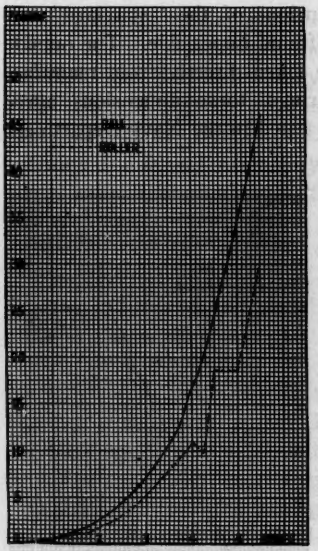

FIG. 3—WEIGHTS IN POUNDS OF "STANDARD SINGLE" BALL AND "SMALL" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

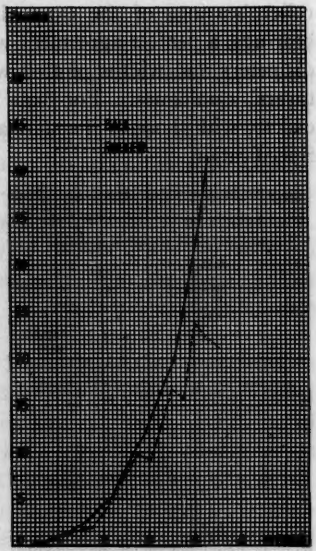

FIG. 4—WEIGHTS IN POUNDS OF "STANDARD DOUBLE" BALL AND "LARGE" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

The weights of "standard" ball bearings are not specified in the Navy Department Specification. The weights of both types were therefore taken from manufacturers' catalogues. They are assumed to be actual and not calculated weights. The table of tapered roller bearing characteristics did not list weights for every bearing. For that reason, the data are not complete but the missing figures are so distributed that it is safe to assume that the curves plotted give approximately correct trends across the gaps.

TABLE I.

CATALOG DATA - SINGLE BALL BEARINGS

BORE ins.	O.D. ins.	WIDTH ins.	WEIGHT lbs.	LOAD CAPACITY - pounds						
				RADIAL						THRUST
				CATALOG CAPACITY(C)	C/2	C/3	C/4	C/5	C/6	% of C
.394	1.378	.433	.11	170	85	57	42	34	28	90
.472	1.457	.472	.12	200	100	67	50	40	33	90
.590	1.654	.512	.15	250	125	83	63	50	42	90
.669	1.850	.551	.27	310	155	103	78	62	52	90
.787	2.047	.591	.34	380	190	127	95	76	63	90
.984	2.441	.669	.55	500	250	167	125	100	83	90
1.181	2.835	.748	.80	575	288	192	144	115	96	90
1.378	3.150	.827	1.03	715	358	238	179	143	119	90
1.575	3.543	.906	1.41	910	455	303	228	182	152	90
1.772	3.937	.984	1.88	1115	558	372	279	223	186	90
1.969	4.331	1.063	2.47	1335	668	445	334	267	223	90
2.165	4.724	1.142	3.07	1585	793	528	396	317	264	90
2.362	5.118	1.220	3.81	1840	920	613	460	368	306	90
2.559	5.512	1.299	4.70	1732	866	577	433	346	289	90
2.756	5.906	1.378	5.72	1932	966	644	483	386	322	90
2.953	6.299	1.457	6.84	2208	1104	736	552	442	368	90
3.150	6.693	1.535	8.13	2466	1234	823	617	494	411	90
3.346	7.087	1.614	9.55	2766	1383	923	692	554	461	90
3.543	7.480	1.693	11.03	3066	1534	1023	767	614	511	90
3.740	7.874	1.772	12.78	3550	1775	850	638	510	425	90
3.937	8.465	1.850	15.86	2796	1398	932	699	559	466	90
4.134	8.858	1.929	18.22	3051	1525	1017	763	610	509	90
4.331	9.449	1.969	21.47	3591	1795	1197	898	718	599	90
4.724	10.236	2.165	29.00	4650	2325	1550	1162	930	775	90
5.118	11.024	2.323	37.00	5000	2500	1667	1250	1000	833	90
5.512	11.811	2.480	46.00	5300	2650	1767	1325	1060	883	90

TABLE II

CATALOG DATA SMALL TAPERED ROLLER BEARINGS

BORE ins.	O.D. ins.	WIDTH ins.	WEIGHT lbs.	LOAD CAPACITY - POUNDS				THRUST
				RADIAL				
				CATALOG CAPACITY(C)	.600	.492	.500	% of C
.475	1.260	.425	.09	38	30	25	19	137
.590	1.378	.433	.11	46	37	30	23	150
.787	1.850	.546	.25	116	93	70	58	120
1.000	2.378	.748	.57	208	168	138	105	110
1.250	2.717	.771	.71	220	176	145	110	127
1.500	3.181	.884	1.01	280	220	225	175	134
1.750	3.346	.854	1.14	285	480	341	263	108
2.000	3.937	.875	1.81	555	4444	341	278	115
2.250	4.250	1.154	2.38	795	636	517	398	118
2.500	4.724	1.148	3.12	935	748	608	468	188
2.750	5.000	1.484	4.09	1255	980	805	618	121
3.000	5.512	1.421	5.12	1295	1036	848	648	135
3.250	6.000	1.625	xx	1905	1524	1236	953	134
3.500	6.535	1.625	xx	2085	1668	1361	1043	137
3.750	6.625	1.625	xx	2085	1668	1361	1043	137
4.000	7.125	1.090	10.50	2290	2512	1879	1445	128
4.250	7.500	1.938	9.80	3355	2684	2181	1678	156
4.500	9.000	1.946	18.55	4600	3680	2990	2300	248
4.750	9.250	2.500	xx	6810	5448	4427	3405	183
5.000	9.000	1.946	18.43	4600	3680	2990	2300	245
5.250	9.250	2.500	xx	6810	5448	4427	3405	183
5.500	10.000	2.625	29.60	7565	6052	4917	3783	136

The data for weights are tabulated in column 4 of Tables I, II, III and IV. These data are plotted against bore diameter in Figures 3 and 4.

TABLE III.

CATALOG DATA DOUBLE BALL BEARINGS

BORE in.	O.D. in.	WIDTH in.	WEIGHT lbs.	LOAD CAPACITY - POUNDS						THRUST % of C.
					RADIAL					
				CATALOG CAPACITY(C)	C/2	C/3	C/4	C/5	C/6	
.472	1.457	.750	.20	125	73	48	36	29	24	90
.590	1.654	.750	.25	200	140	93	70	56	47	90
.669	1.850	.875	.31	350	205	177	133	106	88	90
.787	2.047	.875	.50	650	325	217	163	130	108	90
.984	2.441	1.000	.75	850	425	283	213	170	142	90
1.181	2.836	1.188	1.19	985	493	328	246	197	164	90
1.378	3.150	1.375	1.75	1215	608	405	304	243	203	90
1.378	3.543	1.438	2.31	1540	770	513	385	308	257	90
1.772	3.937	1.563	3.13	1915	958	638	479	383	318	90
1.969	4.331	1.750	4.25	2265	1133	755	566	453	378	90
2.165	4.724	1.938	5.87	2700	1350	900	675	540	450	90
2.362	5.118	2.125	6.94	3185	1593	1062	798	637	531	90
2.559	5.512	2.313	8.61	3938	1466	977	733	586	489	90
2.756	5.906	2.500	10.75	3686	1844	1096	822	658	548	90
2.953	6.299	2.688	12.13	3752	1876	1251	938	750	625	90
3.150	6.693	2.688	15.00	4800	2400	1600	1056	840	700	90
3.346	7.087	2.875	17.50	4625	2313	1542	1156	925	771	90
3.543	7.480	2.875	19.70	5200	2600	1733	1300	1040	867	90
3.740	7.874	3.063	24.70	4350	2175	1450	1088	870	725	90
3.937	8.465	3.250	29.50	4750	2375	1583	1188	950	792	90
4.134	8.858	3.438	34.50	5100	2550	1700	1275	1020	850	90
4.331	9.449	3.625	41.50	6100	3050	2033	1525	1220	1017	90

TABLE IV

CATALOG DATA LARGE TAPERED ROLLER BEARINGS.

BORE in.	O.D. in.	WIDTH in.	WEIGHT lbs.	LOAD CAPACITY - Pounds.					THRUST % of C.
					RADIAL				
				CATALOG CAPACITY(C)	.800	.650	.500		
.472	1.260	.425	.09	58	30	25	19		137
.590	1.378	.435	.11	44	37	30	23		150
.750	1.938	.688	.44	160	128	104	80		87.5
1.000	2.515	1.001	.94	345	276	224	173		84
1.250	2.859	1.161	1.26	470	376	306	235		112
1.500	3.484	1.145	1.76	680	520	425	325		88.5
1.750	3.750	1.177	1.97	700	560	455	350		95
2.000	4.375	1.455	3.65	1040	832	676	520		99
2.250	4.750	1.625	xx	1550	1284	995	765		105
2.500	5.345	2.205	7.01	2610	2088	1697	1305		113
2.750	5.875	2.155	9.77	2970	2376	1931	1485		121
3.000	5.875	2.155	9.11	2970	2376	1931	1485		121
3.250	6.625	2.219	12.54	4130	3304	2686	2065		100
3.500	7.500	2.265	14.60	4655	3724	3026	2328		112
3.750	7.500	2.265	15.76	4655	3724	3026	2328		112
4.000	8.375	2.625	25.75	6045	4836	3929	3023		109
4.250	8.375	2.625	22.56	6045	4836	3929	3023		109
4.500	8.375	2.625	21.15	6045	4836	3929	3023		109

DIMENSIONS—" SINGLE" BALL AND " SMALL" TAPERED ROLLER BEARINGS. FIGURE 1.

Figure 1 shows a distinct advantage for " small" roller bearings in outside diameter, that is the outside diameter is less than that of the " single" ball bearing of equal bore. In width, it is seen that, as plotted, the graph of ball bearing widths is very close to being the smoothed curve of the roller bearing widths. It can be assumed from this Figure that if a series of roller bearings were built to compete with the " standard" ball bearings, the bores and widths would be practically equal in the two types, and the roller bearing would have a considerably smaller outside diameter.

DIMENSIONS—" DOUBLE" BALL AND " LARGE" TAPERED ROLLER BEARINGS. FIGURE 2.

Figure 2 shows an advantage in both width and outside diameter of " large" roller over " double" ball bearings. In this case, the difference is slight in outside diameter but very appreciable in width except for the smaller size. This is so, of course, because the " double" row ball bearing is necessarily wider than the " single" row.

WEIGHT—" SINGLE" BALL AND " SMALL" TAPERED ROLLER BEARINGS. FIGURE 3.

As would be expected from the previous comparison of dimensions, the weights of " small" roller bearings are consistently less than those of the " single" ball bearings.

WEIGHT—" DOUBLE" BALL AND " LARGE" TAPERED ROLLER BEARINGS. FIGURE 4.

In general, Figure 4 shows an advantage in weight of roller over ball bearings. This advantage is not as consistent as for the lighter weight series. It will be noted that between 1¼ and ¾ inches and between 2⅜ and 2⅛ inches the " double" ball bearing is lighter than the " large" roller.

RADIAL LOAD CAPACITY. FIGURES 5, 6, 7 AND 8.

A comparison of radial load capacities, based on catalogue figures is not simple. In determining capacities of bearings, each manufacturer uses methods of his own, known only to himself.

He lists "catalogue" capacities along with instructions for modifying these capacities for various applications. In comparing the products of any two manufacturers, the differences in the listed capacities and the factors of applicability can easily be made to show an advantage for either one or the other as desired. However, as this paper is intended to give an impartial comparison, considerable thought was given to the best method of comparison.

The roller bearing manufacturer in his catalogue uses four "multipliers": 1.0, 0.80, 0.65, and 0.50 and describes the applications where each should be used as follows:

100 per cent rating.	General machinery applications, such as gear boxes of automobiles, tractors, road rollers, cranes, cement mixing machines, machine tools, and similar applications.
80 per cent rating.	Trailer wheels on solid tires. Wheel applications on iron and steel tires operating on steel rails. Heavy machinery applications up to 1000 R.P.M., 24-hour service, etc.
65 per cent rating.	Crawler type tractor idler wheel mechanisms. Applications subject to extreme shock such as internal combustion engine crankshafts, air compressor crankshafts, oil well machinery, excavating shovels. High speed machinery applications at 1000 R.P.M. and above on continuous duty, 24 hours per day. Centrifugal pumps, blowers, fans, etc.
50 per cent rating.	Differential axles for automobiles.

It is apparent that in selecting a multiplier to use in determining the load capacity of a roller bearing for a particular application, that application must be fitted into its proper place in the above table. A study of the four classes of machinery above shows almost unmistakably that *general* Naval machinery would fall into the third class which requires a 65 per cent bearing rating. If anything, the requirements for bearings in this class are more severe than the general Naval requirement but this is on the side

of greater safety. This classification is for machinery of the following general characteristics:

a. Subject to extreme shock.
b. 1000 R.P.M. and greater.
c. Continuous duty, 24 hours per day.

Specifically listed under the class are: air compressor crankshafts, centrifugal pumps, blowers, and fans, all of which are units of Naval machinery.

For *general* Naval applications 0.65 will be accepted as the proper "multiplier" for radial load capacities listed in the catalogue. It should be understood, however, that this has been done merely to generalize the comparisons to be made and that for any particular application a more detailed study for determination of the correct "multiplier" should be made.

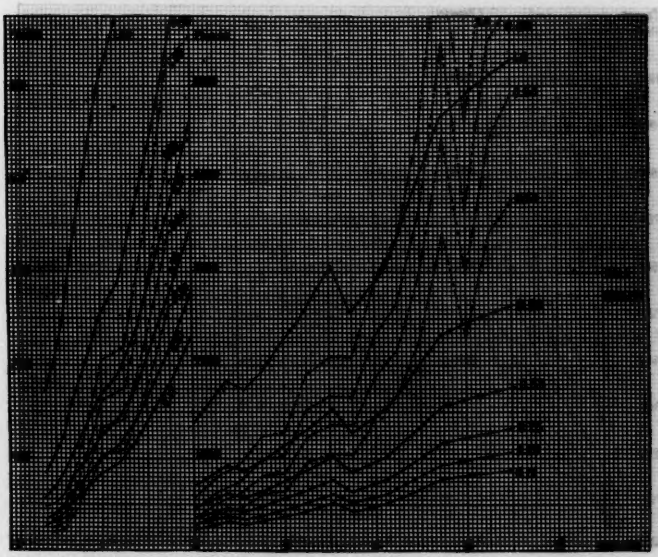

FIG. 5—RADIAL LOAD CAPACITIES WITH VARIOUS MULTIPLIERS OF "STANDARD SINGLE" BALL AND "SMALL" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

The radial load capacities for the "small" and "large" series of tapered roller bearings are given in columns 5, 6, 7 and 8 of Tables II and IV. The column headed "Catalogue Capacity (C)" gives the capacity which is listed in the catalogue for the bearing at 1000 R.P.M. In the catalogue radial load capacities are listed for a number of speeds, 1000 R.P.M. being arbitrarily selected for the purpose of the comparison now to be made. Selection of this one speed does not affect the application of the "multipliers." Columns headed .80C, .65C, and .50C in Tables II and IV give the capacities as reduced by use of the "multipliers" at the head of each, *e.g.*, .80C indicates 80 per cent of the catalogue capacity given in column 5.

The data given in columns 5, 6, 7 and 8 are plotted in broken lines on Figures 5 and 6.

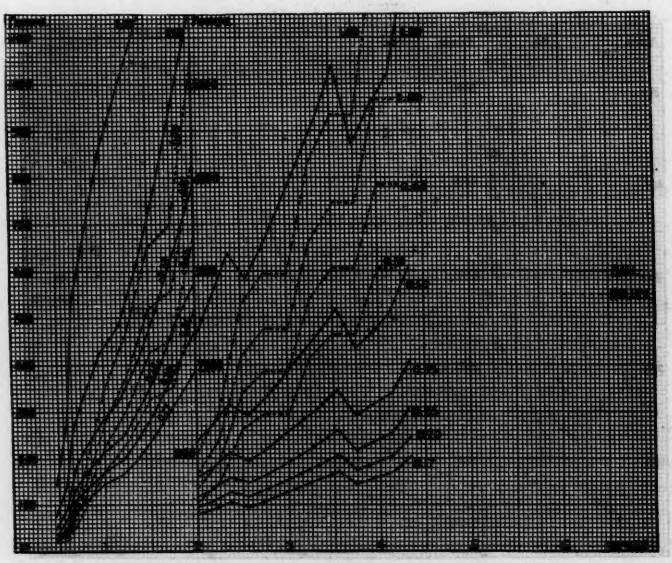

FIG. 6—RADIAL LOAD CAPACITIES WITH VARIOUS MULTIPLIERS OF "STANDARD DOUBLE" BALL AND "LARGE" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

The ball bearing manufacturer gives four factors, each of which varies with the application, the product of the four, for any application, being the reciprocal of the "multiplier" which is to be applied to the catalogue capacity. These factors are given and defined below:

"A"—TYPE OF SERVICE.	*Factor*
Intermittent and seasonal	0.5
Average shop production (3000 hours per year)	1.0
Heavy production, average life	2.0
Continuous production, long life	3.0

"B"—TYPE OF LOAD.	
Uniform and steady	1.0
Variable and light shock	1.5
Shock and vibration	2.0
Severe shock	3.0

"C"—CATALOGUE FACTOR.

Bearing Size.	*Under 100 R.P.M.*	*100 to 2000 R.P.M.*	*Above 2000 R.P.M.*
00-05	0.8	1.0	0.8
06-12	1.0	1.2	1.0
13-18	1.2	1.5	1.2
19 and larger	1.5	2.0	1.5

"D"—TYPE OF INSTALLATION.	
Well established design practice	1.0
Approximate or arbitrary design practice	1.5
Indeterminate load or special hazard	2.0

It is seen that the product of these factors may run as high as 36, giving a multiplier of 0.028, and would rarely, if ever, be less than 2, giving a multiplier of 0.50. It is therefore to be expected that the *catalogue* radial load capacities of ball bearings will be greater than the *catalogue* radial load capacities of roller bearings.

Factor "C" varies with the size of bearing and with speed. For the purposes of this comparison it can therefore be applied exactly. This has been done in Tables I and III where the figures listed as "Catalogue Capacities" are actually the listed catalogue capacities reduced in each case by the factor "C" proper for the bearing size and 1000 R.P.M.

The final factor used here, the product of only the three separate factors, "A," "B" and "D," is the reciprocal of the "multiplier" which must be applied to the "catalogue" capacities given in Tables I and III. The "A" × "B" × "D" factors which are obtainable from the tabulation given above and the resultant "multipliers" are:

Factor.	*Multiplier.*	*Factor*	*Multiplier.*
½	2.00	4½	.22
¾	1.33	6	.17
1	1.00	6¾	.15
1⅛	.89	8	.13
1½	.67	9	.11
2	.50	12	.08
2¼	.44	13½	.07
3	.33	18	.06
4	.25		

To select a factor from the above list and to state that the factor so selected represents the proper one for *general* Naval machinery applications and also that it is comparable to the 0.65 multiplier already selected for roller bearings is not possible without some chance of error. If for factor "A," 3.0 is chosen for continuous production and long life; and for factor "B" 3.0 is chosen for severe shock, the resultant "multiplier" is 0.11. Design is not mentioned under roller bearing "multiplier" value so factor "D" cannot be selected as a comparative value. This method of selection considers the terminology of the two manufacturers as the same. If, however, the discussion on selection of roller bearing "multiplier" is forgotten for the time and factors are selected for ball bearings with nothing in mind but *general* Naval application, it would seem that the factors which should be selected would be:

"A"—Heavy production, average life	2.0
"B"—Shock and vibration	2.0
"D"—Approximate or arbitrary design practice	2.0
Final factor	8.0
"Multiplier"	.125

An argument which might now be advanced, and not without merit, is that for a general or average case it might be expected that, of the three factors, one might easily be expected to be less

than 2.0, probably only 1.0. If this argument is accepted, the factor becomes 4.0 with a "multiplier" of 0.25. There is the possibility that this multiplier, when used for comparing the two types of bearings, may be somewhat favorable to the ball bearings. However, based on the discussion given, 0.25 has been selected as the correct "multiplier" for catalogue load capacities for ball bearings and will be used for comparison.

A complete tabulation of all the possible capacities would require fourteen columns. This completeness is unnecessary and undesirable. In Tables I and III various capacities are listed in columns 5, 6, 7, 8, 9 and 10. For simplicity the factors used were 1, 2, 3, 4, 5 and 6, permitting approximate interpolation throughout the range of probable actual factors. The smaller and larger factors would be applicable only in very special cases. As previously stated the value listed as "Catalogue capacity (C)" is not

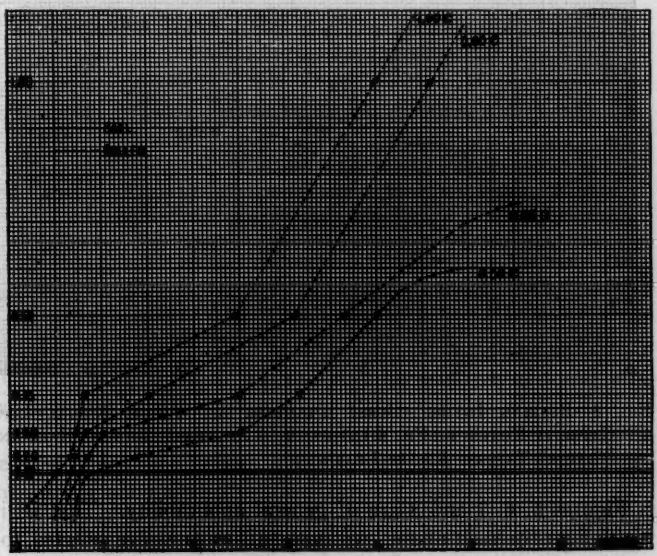

FIG. 7—BEARING INSIDE (BORE) DIAMETER AT WHICH RADIAL LOAD CAPACITY CURVES OF FIG. 5 INTERSECT—"STANDARD SINGLE" BALL AND "SMALL" TAPERED ROLLER BEARINGS.

the actual value given in the catalogue but is this value multiplied by the reciprocal of factor "C" for 1000 R.P.M. and the size of each bearing.

The data given in columns 5 to 10, inclusive, of Tables I and III, are plotted in solid lines on Figures 5 and 6.

Figures 7 and 8 were plotted from the data given in Table V. These data give the bearing sizes (bore diameter) at which each capacity curve for ball bearings intersects each capacity curve for roller bearings.

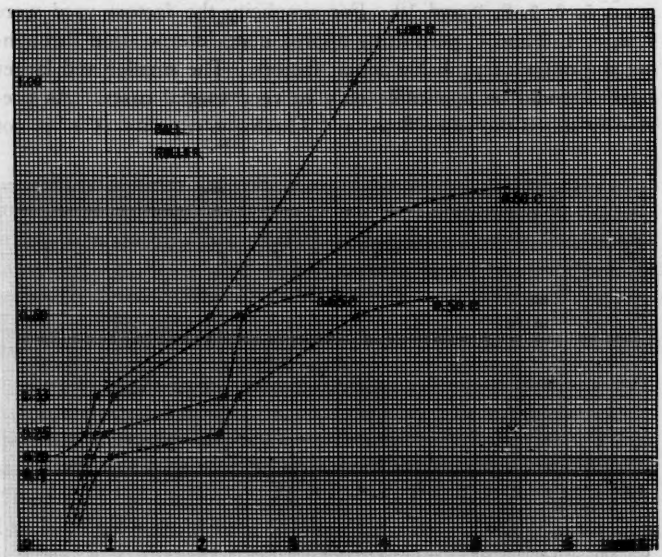

FIG. 8—BEARING INSIDE (BORE) DIAMETER AT WHICH RADIAL LOAD CAPACITY CURVES OF FIG. 6 INTERSECT—"STANDARD DOUBLE" BALL AND "LARGE" TAPERED ROLLER BEARINGS.

Figures 5, 6, 7 and 8 are of general applicability. That is, if multipliers for each of the two types of bearings are selected for a particular machine or application, Figures 5 and 6 will show which of the types will give the better load characteristics for the size desired and Figures 7 and 8 will show over what range of

TABLE V

BORE DIAMETERS AT POINTS OF INTERSECTION OF LOAD CAPACITY CURVES

A. SINGLE BALL - SMALL ROLLER.

BALL BEARING MULTIPLIER	1.00	0.50	0.33	0.25	0.20	0.17
		ROLLER BEARING MULTIPLIER - 1.00				
BORE	4.00	2.45	0.80	0.70	0.65	0.60
		ROLLER BEARING MULTIPLIER - 0.80				
BORE	4.60	3.10	1.50	0.80	0.70	0.65
		ROLLER BEARING MULTIPLIER - 0.65				
BORE	xx	3.65	2.50	1.00	0.83	0.73
		ROLLER BEARING MULTIPLIER - 0.50				
BORE	xx	4.00	3.15	2.50	1.40	0.85

B. DOUBLE BALL - LARGE ROLLER.

BALL BEARING MULTIPLIER	1.00	0.50	0.33	0.25	0.20	0.17
		ROLLER BEARING MULTIPLIER - 1.00				
BORE	3.70	2.10	0.85	0.75	xx	xx
		ROLLER BEARING MULTIPLIER - 0.80				
BORE	xx	2.37	1.07	0.83	0.75	0.70
		ROLLER BEARING MULTIPLIER - 0.65				
BORE	xx	2.45	2.25	0.95	0.82	0.75
		ROLLER BEARING MULTIPLIER - 0.50				
BORE	xx	3.66	3.40	2.20	1.00	0.85

sizes one type will be preferable. These figures as given are of a handbook nature. That is, they can be used by a designer for selecting the better type of bearing after he knows all the details of his application which will influence his selection of radial load capacity multipliers.

Only those portions of Figures 5, 6, 7 and 8 which show the selected *general* multipliers, 0.65 for roller and 0.25 for ball bearings, are actually needed for the following discussion.

"SINGLE" BALL AND "SMALL" TAPERED ROLLER BEARINGS. FIGURES 5 AND 7.

Figure 5 is available but need not be used. Figure 7 shows that regardless of the multiplier used, the "small" tapered roller bearing will have a greater radial load capacity than the "single"

ball bearing in all sizes greater than 4.6 inches and that the "single" ball will have a greater capacity in all sizes below 0.5 inches. This is so because beyond these points no intersections occur. In the intermediate range the type which will have the advantage will depend on the comparable multiplier. Using those already selected for our purposes, 0.65 for tapered roller bearings and 0.25 for ball bearings, it is seen that the 0.65C broken line intersects the 0.25C full line at a bore diameter of 1.0 inch and at this point the broken line rises continuously to the right. It is apparent, then, that the "small" tapered roller bearing has greater radial load capacities than has the "single" ball bearing for all sizes of bearings of 1 inch bore diameter or greater. It is also apparent that for all sizes below 1 inch, the "single" ball has the greater capacity.

"DOUBLE" BALL AND "LARGE" TAPERED ROLLER BEARINGS. FIGURES 6 AND 8.

Qualitatively, Figures 6 and 8 are similar respectively to Figures 5 and 7. Quantitatively there is a slight difference. Figure 8 shows general fields of greater load capacity only by eliminating multipliers of 1.0. If this is done it is seen that the "large" tapered roller bearing has the greater radial load capacity in all sizes greater than 3.70 inches and the "double" ball bearing has the greater capacity in all sizes below 0.7 inches. The assumption which has been made is reasonable because it merely means, as a study of the figure will show, that a multiplier of 0.50 for roller bearings would never be comparable to one greater than 0.50 for ball bearings and that a multiplier of 0.167 for ball bearings would never be comparable to one greater than 0.80 for roller bearings.

On Figure 8 the broken line 0.65C intersects the full line 0.25C at a little less than 1 inch bore diameter. It therefore appears that the "large" tapered roller bearings have a greater radial load capacity than have the "double" ball bearings in all sizes greater than 0.95 inches and that the advantage shifts to the ball at sizes less than 0.95 inches.

THRUST LOAD. FIGURES 9 AND 10.

The thrust load capacities of roller bearings have been taken directly from the catalogue and then expressed as percentages of

the radial load capacities. The percentages are given in the last columns of Tables II and IV and are plotted in broken lines on Figures 9 and 10.

The most recent catalogue of the ball bearing manufacturer which was available states that for thrust capacities the engineering department of the manufacturer should be consulted. However, a previous catalogue states that the thrust capacity of ball bearings

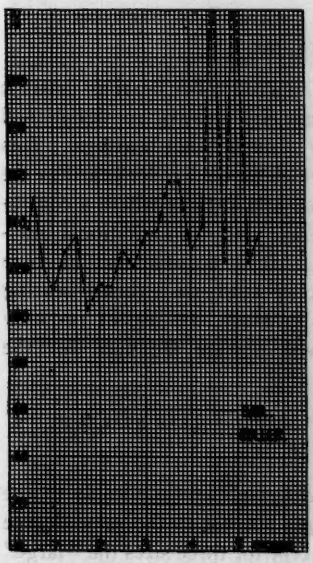

FIG. 9—THRUST LOAD CAPACITIES EXPRESSED AS PERCENTAGES OF RADIAL LOAD CAPACITIES OF "STANDARD SINGLE" BALL AND "SMALL" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

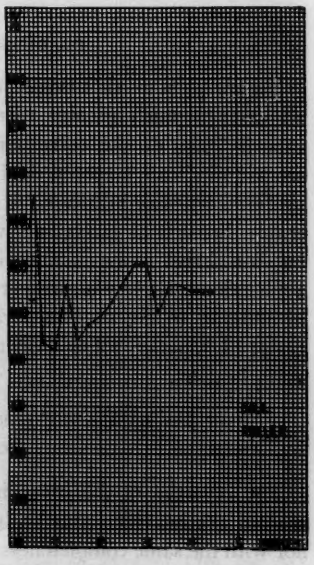

FIG. 10—THRUST LOAD CAPACITIES EXPRESSED AS PERCENTAGES OF RADIAL LOAD CAPACITIES OF "STANDARD DOUBLE" BALL AND "LARGE" TAPERED ROLLER BEARINGS VS. BEARING INSIDE (BORE) DIAMETERS.

at 1000 R.P.M. is 90 per cent of the radial load capacity given for that speed. Though constant, this percentage is listed in the last column of Tables I and III for completeness and is plotted in solid lines on Figures 9 and 10.

The broken lines in Figures 9 and 10 are decidedly irregular. This can be accounted for by the fact that the tapered roller bearings selected for the "small" and "large" series were not designed as a regular series but were each individually designed for a particular application. It is probable that some of these applications required a much greater comparative thrust capacity than did others and that this greater thrust capacity was provided in the detailed design by a variation in the taper of the rollers.

"SINGLE" BALL BEARINGS AND "SMALL" TAPERED ROLLER BEARINGS. FIGURE 9.

Figure 9 shows that the "small" tapered roller bearings have in all cases a much greater thrust capacity relative to radial load capacity than do the corresponding "single" ball bearings. This seems to indicate a special field in which tapered roller bearings are preferable to ball bearings, viz., those applications where a considerable thrust load exists and where the use of an additional bearing for thrust alone is undesirable. It is possible that in some applications where the present practice is to use a thrust bearing in addition to the necessary radial bearings, the use of properly designed tapered roller bearings will obviate the need for the extra bearing.

"DOUBLE" BALL AND "LARGE" TAPERED ROLLER BEARINGS. FIGURE 10.

The remarks in the preceding paragraph are applicable to the case of the heavier series of bearings but not to the same degree nor with the same completeness. Except for three sizes the "large" tapered roller bearing has an advantage in relative thrust capacity over "double" ball bearings. These three exceptions are all in sizes of 1½ inches or smaller.

SPEED. FIGURE 11.

Figure 11 shows a comparison between the effects of speeds on the life of the two types of bearings. The factor shown plotted against speed is that multiplier of the life of the bearing which would be obtained by running it fully loaded at the speed shown rather than at 1000 R.P.M. It can also be used to show what load

can be carried to give the same bearing life at some speed other than 1000 R.P.M. As an example: Suppose that the curve shows a factor of 0.80 for a speed of 1500 R.P.M. This indicates that a bearing whose load capacity at 1000 R.P.M. is "L" pounds for a life of "T" hours, will have a life of 0.80T hours if run at 1500 R.P.M. with the load L or that it can carry a load of 0.80L at 1500 R.P.M. for a life of T hours.

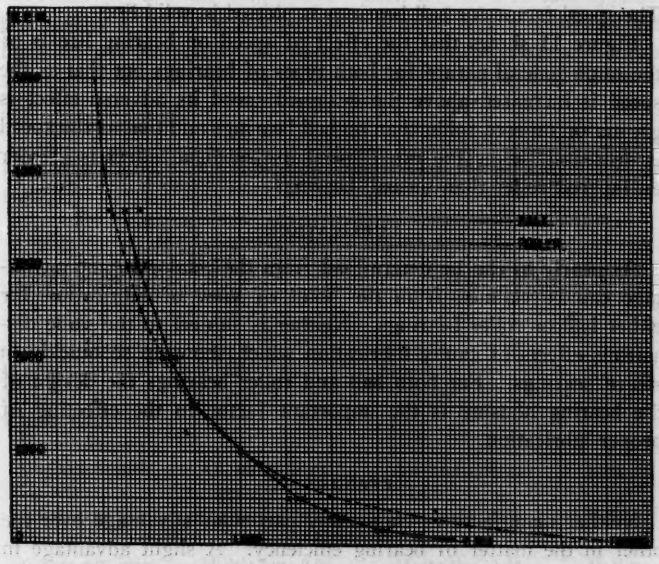

FIG. 11—SPEED FACTORS APPLICABLE TO LOAD OR BEARING LIFE FOR ALL BALL AND FOR ALL TAPERED ROLLER BEARINGS.

The speed factors for tapered roller bearings, applicable to either the "small" or the "large" series, are taken directly from a table published in the catalogue. As listed in the catalogue, unity is taken for a speed of 500 R.P.M. so a shift was required to move unity to 1000 R.P.M. These data are plotted in broken lines on Figure 11.

The ball bearing catalogue does not give a similar table but the radial load capacity of each bearing is listed for a number of different speeds. A number of bearings were selected at random

from both the "single" and "double" series and the speed factor calculated for each speed for which the radial load capacity was given. Thus the radial load capacity of a bearing at 2000 R.P.M. divided by the capacity of the same bearing at 1000 R.P.M. is the speed factor at 2000 R.P.M. All the factors calculated are marked on Figure 11 with small crosses. A smooth curve drawn through the median points at each speed is very regular and readily permits a comparison with the speed factor of tapered roller bearings. The speed factor for ball bearings is plotted in solid line.

Figure 11 shows that the effect of speed on bearing life is of the same general nature for both types of bearings but that as the speed is increased above the designed speed its adverse effect is greater on tapered roller than on ball bearings and that as the speed is decreased below the rated speed its beneficial effect is greater on tapered roller than on ball bearings.

LUBRICATION.

As nearly as can be determined from the catalogues, no advantage can be claimed from the point of view of lubrication for either ball or roller bearings. Neither is subject to the same disastrous effects of faulty lubrication which show up so quickly in sleeve bearings. For both ball and roller bearings the design of the bearing housing is the important factor in the effecting of proper lubrication.

EFFICIENCY.

Neither manufacturer claims any particular advantage over the other in the matter of bearing efficiency. A slight advantage in coefficient of friction is claimed and admitted for the ball bearing but it is not large. This advantage is admitted only for laboratory determinations and not for actual service determinations. Since both types of bearings utilize rolling contacts, the ball a point and the roller a line, and the greater part of the pure frictional losses is in the cages of both types, there appears to be no reason why the two should vary greatly in efficiency.

CONCLUSIONS.

It was stated that the results desired were answers to the questions:

1. Are ball and tapered roller bearings competitive in application?

2. Are there two different fields of application, one for ball and one for tapered roller bearings?

The questions can be answered in two different ways. Both types of bearings are "anti-friction" bearings and nothing was brought out in a study of the catalogues which would indicate that the two types are not competitive. That is: for a certain application, either a ball or a tapered roller bearing can be designed and installed and the results will be satisfactory in both cases. In this sense the two types are competitive, and the answers to the questions are, to question 1, yes; to question 2, no.

Considering the quantitative design and operational features the following table summarizes the results of the previous discussion:

SINGLE BALL VERSUS SMALL ROLLER.

Point of Comparison	*Single Ball*	*Small Roller*	*More Advantageous*
Outside Diameter	Larger	Smaller	Roller
Width	Equal	Equal	Neither
Weight	Heavier	Lighter	Roller
Radial Load Capacities			
0 to 0.5 inch	All multipliers greater	All multipliers less	Ball
0.5 to 0.95 inch	Multiplier 0.25 greater	Multiplier 0.65 less	Ball
0.95 to 4.0 inch	Multiplier 0.25 less	Multiplier 0.65 greater	Roller
4.0 inches and greater	All multipliers less	All multipliers greater	Roller
Thrust/Radial Load Capacity	Less	Greater	Roller
Increased Speed Above Designed Speed	Less reduction of life	Greater reduction of life	Ball
Decreased Speed Below Designed Speed	Less increase of life	Greater increase of life	Roller

DOUBLE BALL VERSUS LARGE ROLLER.

Point of Comparison	*Double Ball*	*Large Roller*	*More Advantageous*
Outside Diameter	Somewhat larger	Somewhat smaller	Roller
Width	Larger	Smaller	Roller
Weight	Heavier	Lighter	Roller
Radial Load Capacity			
0 to 0.7 inch	Any multiplier greater	Any multiplier less	Ball
0.7 to 1.0 inch	Multiplier 0.25 greater	Multiplier 0.65 less	Ball
1.0 to 3.7 inches	Multiplier 0.25 less	Multiplier 0.65 greater	Roller
3.7 inch and greater	Any multiplier less	Any multiplier greater	Roller
Thrust/Radial Load Capacity	Generally less	Generally greater	Roller
Increased Speed Above Designed Speed	Less reduction of life	Greater reduction of life	Ball
Decreased Speed Below Designed Speed	Less increase of life	Greater increase of life	Roller

From the above tables it appears that the questions can be answered in another way, that is, that the two types are not competitive but have separate and definite fields of application which can be described as follows:

Depending on the load, "double" and "single" ball bearings should be used in sizes below one inch in cases where the thrust load relative to the radial is not large and in cases where the tendency will be to operate at higher rather than lower speeds than the designed speed.

Depending on the load, "large" and "small" tapered roller bearings should be used in sizes of one inch and above in all cases where weight and size are important considerations; in cases where the thrust load relative to the radial load is large; and in cases where the tendency will be to operate at lower rather than higher speeds than the designed speed.

The above indicates the general fields of application of the two types. In practice, the actual design figures must be applied to determine which bearing should be used.

To make the results of this study more conclusive, it is desirable that all the data used in this paper be checked by tests in a non-partisan laboratory. The procedure used in this paper may then be used with any new data made available by such tests to establish more definitely the relative advantages of the two types of bearings.

If the use of anti-friction bearings is indicated for a new design, the relative merits of the two types can easily be determined for that design, using appropriate "multipliers" and following the method indicated above, and the type of bearing which shows more advantages then selected with reasonable assurance.

SOME ENGINEERING EXPERIENCES AFLOAT.

By Lieut. Commander T. A. Solberg, U. S. N., Member.

The following pages have been set down for the twofold purpose of recording some engineering experiences afloat which may be of possible interest, and to detail some of the methods, precepts, and practices which have been found to give dividends in operating results, as well as in upkeep.

The events described occurred during tours of sea duty in Engineering—first, in a small three-thousand-ton cruiser during the war; second, a short tour, including commissioning, in one of the first electric drive battleships; third, three years in an auxiliary vessel; and, recently, a cruise in a first-line battleship.

The experiences narrated are, in many cases, of usual every-day occurrence, but some took place during times of stress, which lifted them out of the ordinary class. Likewise, it will be found that many of the practices and methods described are of ancient vintage—perhaps this accounts for their lack of use in many instances. Some, however, will be found to be of more recent development, and it is hoped that some will be new "wrinkles" to at least a few, while still others may profit by using variations of the practices suggested.

In order to present experiences, methods, and practices in a compact form, the general scheme has been to divide the machinery plant into various units and to discuss all the above in connection with these subdivisions as far as possible. It is naturally impossible to provide a connected story and no effort will be made to do this.

MAIN ENGINES AND APPURTENANCES.

Under the foregoing heading condensing plants are chosen for discussion first, for the reason that troubles with them stand out perhaps above all others. Leaky condensers—main, auxiliary, or both—seem to have been a regular inheritance on every cruise.

In the summer of 1917, just after the U. S. S. *Tacoma* returned from seven months in the tropics, the writer came back from his first liberty in the United States to find himself her Engineer Officer, because of the wholesale detachment of many regular commissioned officers. The ship's duty was to convoy low and medium powered cargo vessels—with an occasional short 14-knot troop convoy interspersed—to the vicinity of the Irish coast, and then return. These trips required from twenty-five to thirty days. As there was no possibility of refuelling en route, a deckload of two hundred and fifty tons of coal was necessary in order to make the nearest port—St. Johns or Halifax.

On the very first trip, the port condenser gave trouble, which gradually increased until retubing was accomplished after the third trip. Leaving port with tight condensers meant a condensate salinity never less than one-half a grain of chlorine per gallon. The condenser ferrules were so badly corroded or "dezincified" that it was useless to try to remove or tighten one, as they invariably snapped off. Bad leaks were plugged with taper plugs. But the constant seep or weep type of leak was the principal cause of trouble. After many occasions when it was necessary at sea to run on one engine while testing the opposite condenser, these were improved materially by sopping the tube sheets with hot linseed oil with a good vacuum on the condenser. This drew oil in around the ferrules and into the deteriorated packing and was the only remedy which helped. In spite of this, it was not unusual to have all the boilers up to 200 grains per gallon during most of the trips, even though both boiler rotation and bottom blow were used to keep the salinity down.

On this same first trip, shortly after leaving the convoy and heading for the general vicinity of St. Johns, the rough weather usual in northern latitudes from October on through the winter, set in. Because of the cold sea water the circulators, which were compound reciprocating driven, were run as slowly as possible for economy. While cleaning fires about 4 P. M., the drop in steam pressure was enough to cause the starboard circulator to stop, unnoticed. When discovered, the vacuum was near zero and the zeal of the oiler in starting the circulator at high speed resulted in such a heavy load on the dependent air pump that the resulting water

hammer bent the crossbar connecting the pump rod of the air pump to the engine linkage. This crossbar was about 6½ inches in diameter at the pump rod connection and was deflected enough to cause the bucket to strike the bottom of the pump casing at every stroke. It was remarkable that the casing had not cracked or broken. Running on one engine in rough weather was not pleasant, nor was dismantling the pump and straightening the crossbeam under these conditions. This being four in the afternoon, the Captain was told the engine would be back in service at midnight. A stuttering blacksmith, of recent Italian extraction, and the only available forge, a small portable affair, played a big role in the straightening of the bent member. Finally, after repeated heatings and straightenings, in which every one had a try at swinging the heaviest sledge on the ship, the crossbar was adjudged straight enough. A little after midnight the starboard engine was again at work. This was fortunate, too, for in the early morning the gathering storm reached its peak and from then on the ship was minus a main topmast. Even a pet dog, supposedly a good sailor, was seasick.

Later on, during the same trip, the same accident was narrowly averted and, as a result, one of the three chief machinist mates aboard was relieved from duty. At the end of the trip he went to the hospital and was found to be in the first stages of a mental derangement. These two occurrences taught the valuable lesson which has been used to advantage ever since—that every piece of machinery has a *slowest safe speed* below which it should not be run, regardless of the economy involved. This speed naturally may vary with conditions.

Engineering competition was not a factor during the war but economy was necessary in order to make the round trip with the extra load of coal on deck. Near the end of the war, when particularly favored by good weather and the biggest excess of coal ever carried, it was possible to make the complete round trip to Hampton Roads. But, on arrival in the Roads, there were less than ten tons of coal left in the starboard reserve bunker—enough for auxiliary purposes until morning when the coal barge arrived. The last day, needless to say, was occupied mostly in nervously watching two small coal piles in the reserve bunkers.

The condensing plant in the U. S. S. *Idaho* not only gave much trouble and many worrisome hours, but also gave much satisfaction, in that the results accomplished in improving its auxiliaries gave marked improvement in overall plant economy.

From the very first, condenser troubles were experienced on this ship. One dynamo condenser went rapidly from fair to worse, in in which latter state two to four tube failures each week were normal. The failed tubes for a distance of about six inches from their ends were pock-marked and pitted, which led to the suspicion that this condition might be caused by electrolytic action. These conditions were relieved, largely, by remounting the zincs on long monel studs so as to bring them nearer the tube sheet. This insured bringing the zincs more effectivly into the electrolytic circuit and also kept them continuously protective, because in this position in the high velocity portion of the water stream, the products of corrosion were being washed off continuously. In other words, the zinc surface did not become polarized by a coating of corrosion products. It is not unusual to find the zincs placed close to the manhole plate out of the main water stream and in this position they invariably become ineffective by polarization after a short period. This change, suggested by a member of the Pacific Coast Section of the Board of Inspection and Survey, reduced the tube failures to an average of one a month and for this reason the zincs of all condensers were arranged in this way. It is believed that not only should the zincs be near the tube sheet but that they should be distributed uniformly with relation to the tube sheet area in order to decrease the possibility of local action at any point which otherwise would be remote from any zincs. Many arrangements do not provide uniform distribution of the zincs. This, however, is not a "cure" for all cases of condenser tube corrosion. The port main unit was struggling along valiantly with its original tubes, at that time more than ten years old. On the winter cruise in the tropics, frequent failures occurred, but this condition was improved by shock testing and partial retubing by the Navy Yard.

Not long after completing the first Navy Yard overhaul, the starboard condenser made itself known by occasional failures, but, because of impending modernization, nothing could be done to cor-

rect it at the next two Yard overhauls. On the last winter cruise in the tropics, the corrosion conditions were aecelerated and failures all but wore down both officer and enlisted personnel. Being in fleet formation made the conditions even more trying as the condenser arrangement was such that only the engines on one side could be used when a main condenser was disabled. For this reason repairs were always undertaken after the day's maneuvers were completed. This often required allowing the boilers to salt up somewhat when a leak appeared in the morning.

Before going into a condenser at sea, it was made the practice to run the circulating pump at full speed for ten minutes, in order to break down any tubes which were on the verge of failing. This routine was adopted when, on the occasion of completing one repair, three more tubes failed within a half-hour. Also, the type of failure that was taking place resulted from deep pits, and breaking down any such areas by this excess pressure insured freedom from leaks for at least a breathing spell. Two incidents in connection with these condenser failures stand out and will long be remembered. While making a test on the starboard unit, being underway on the port engines, the port condenser indicated a leak just as the starboard engines were about to be placed in operation again. This necessitated opening up the port condenser as soon as the starboard side was ready. On another occasion, after plugging twelve leaking tubes on the starboard side, as the starboard engines were being warmed up again, a new leak was indicated. The whole process was carried out again and five more tubes plugged. The double maneuvers each required about seven hours of hard labor under the hot humid conditions which usually exist in an engine room in the tropics. Each man of the repair crew probably lost more weight at these times than any football player in the hardest game of the season. And it is at such times that you are prone to think the engineer of coal burning days was right when he said, " Engineering in the Navy is no job for a gentleman."

On one occasion, the rapid salting up of the boilers from a heavy leak in the dynamo condenser gave some unlooked-for experience with foaming boilers. While getting up anchor with the Fleet, leaving Seattle for San Francisco, the after dynamo con-

denser, which was serving the extra generator required for the anchor windlass, had a tube failure and, in an incredibly short time, salted up everything. The leak was reported and the plant secured, but the damage had been done. At two-thirds speed, getting underway, the main condenser condensate showed three grains of chlorine per gallon. About the same time, the forward dynamo room reported salty condensate. Apparently all condensers were leaking and, with both main condensers known to be bad actors, the situation was rather unpleasant. It was determined finally, however, that the boilers were foaming from high salt concentration. Each boiler was given a bottom blow but this had little beneficial effect and it was necessary to go easy on the water available. Two fresh boilers were put on the line, to replace the two with the highest concentration. Twelve hours later these were emptied, refilled with fresh water and again put in service, and one additional boiler was added at the same time. This left only one salty unit on the line and it was operated from then on at the lowest possible rate, in order to prevent the possibility of its carrying over and contaminating the fresh boilers. At one stage of this incident, after nightfall, conditions were such that it was contemplated falling out of formation until conditions could be corrected. All in all, it was a hectic and sleepless twenty-four hours.

As regards general fuel economy, the condensing plant is a vital part of the installation, not only because the engine efficiency is dependent on its proper functioning, but also because the auxiliaries which serve the condenser are often responsible for many wasted B. T. U.'s. The work which was done in connection with the *Idaho's* condenser installation embodies most points of value that might arise and the following applies to that vessel.

Air ejectors for main condensers.—The two main condensers were equipped with a Parson's Augmentor condenser which was designed probably about 1915 and was the earliest development in equipment for attaining high vacuum. The air ejectors were of the single stage type, discharging through the augmentor condenser into the main air pump suction. A look at one of the nozzles during routine overhaul led to the suspicion, which was confirmed by computation, that the steam nozzle was most inefficient.

Some new nozzles were designed and made on board and then tried out as opportunities were presented. It was found that

there was very little information available, either in text books or technical magazines, on the subject of complete ejector design. The problem was complicated by the fact that the new nozzles had to be designed or built around the old equipment or piping. By trial, error, and computations, in which Napier's formula was the principal weapon, new nozzles were evolved which, by changing the original nozzle location in the air suction pipe, gave excellent results. The steam consumption of the new units was slightly less than one-half that of the old jets. At low and moderate speeds, this saving was a sizable percentage of the main engine water rates. This also illustrates the possibilities of taking advantage of new advances in engineering and improving old design at practically no expense and without purchasing complete new equipment. It should be unnecessary to add that the Bureau should be consulted first, both for information and authority.

Another point in connection with any air ejector performance is the run of the steam piping. In this case, the connection to the auxiliary steam line was by means of a fitting in the bottom of the line. In other words, this piping acted as a drain line for this portion of the auxiliary line and, if there was any moisture present, the steam nozzles were certain to get it. The fitting in the steam line was turned over so that the steam connection was at the top and this insured the air ejectors' getting the driest steam available. A steam nozzle falls off rapidly in efficiency of performance with decreasing steam dryness.

Main Circulators.—During a complete and well-deserved overhaul of the main circulators, it was noted that the turbines had seven steam nozzles, of which three were controlled by the main throttle and four had individual valves. Individual valves were installed on two of the throttle controlled valves, so that when conditions were such as to require only one or two nozzles, only that number need be used. As a matter of interest—and instruction for junior officers—actual test runs were made on each circulator to determine the speed, steam chest pressure, and steam consumption characteristics under each nozzle condition. From these curves, it was found that savings of fifteen to thirty per cent of the former circulator water rates were made when operating on the one or two nozzle settings. As ship speeds up to 14 knots were possible on

the one and two nozzle combinations, with sea water at 70 degrees, it can be seen that the installation of the additional nozzles resulted in sizable savings during most underway conditions. Also, the installation of steam gages on the circulator turbine steam chest was found to be conducive to economy, as it made speed regulation both faster and more accurate. No provision had been made for this in the original design.

Main Condenser Cleanliness.—It is not unusual, especially on older ships, to find the steam side of the condensers quite dirty. Generally, the top tubes are the worst and the grease-impacted scale, rust and dirt are most difficult to remove satisfactorily. In one case, much effort was found necessary in order to get even fair results. After diligently boiling out and reboiling with negligible results, it was decided to use some of the same mixture used for washing out fuel oil tanks, as the ingredients were quite harmless. In addition to this, small scrubbing brushes were cut down and secured on handles, so that the brushes could be inserted in the vertical lanes between tubes. These were worked back and forth and thereby much dirt in the upper third of the condenser was removed. Tube interference prevented cleaning the lower tubes in this way. When boiling out, the solution was kept circulating by using a portable pump connected to take suction from the bottom of the condenser and discharge to the top of the condenser. The exhaust trunk was so arranged that the boiling solution level could only cover the lower half of the condenser. To overcome this a spray system was rigged up so that the pump discharged the boiling solution over the top tubes, whence it trickled down over the remaining uncovered tubes. Despite all the trouble taken, results were hardly satisfactory. It is believed that the system for boiling out now prescribed is susceptible of great improvement.

Shafting.—During the cruise in the *Idaho* some interesting observations were made in regard to vibration of the after hull and its causes. They are now of further interest in view of subsequent development of failure in one of the stern tube shafts. On the first winter cruise the ship was in the nine to twelve month out of dock period, although the usual docking occurs at nine months. Immediately on leaving San Pedro with the Fleet the vibration of

the after hull at clean bottom revolution speeds of twelve knots and above was rather severe, and became progressively worse as the time out of dock increased. The trip South was made at engine speeds of 13-14 knots, which happened to be the critical range for the vibration. The noise was so great that conversation across the table in the Ward Room was rather difficult—and sleep was also not as easy as usual A check of the wood clearances reported at the last docking showed that there were four bearings near the tolerance. It was surmised that these large clearances in combination with the foul bottom were responsible for setting up the vibrations.

It was surprising to find on docking at Hunter's Point at the end of the cruise, just twelve months out of dock, that the suspected bearings were still nineteen thousandths within the tolerance limit (239 thousandths). Rewooding was denied by the Navy Yard and the radio hurriedly presented the ship's plea for consideration to the Bureau as it was felt that the same condition would reappear as soon as the bottom became a little foul. The refusal came back just two hours before scheduled undocking. The resourceful Executive Officer, who held the purse strings of the Ship's Service Store, suggested using the long distance telephone to Washington. This was done and the Bureau's decision that rewooding was unnecessary before next docking was finally reversed. The Bureau was of the opinion that the vibration must come from within the ship. The only other source possible would be the engines themselves and the Bureau was informed that the engines were not responsible and if they had been would never have been operated in such a condition.

The subsequent winter cruise when the ship was again forced to go through an engineering year with a twelve month docking interval strengthened the supposition that combined excessive clearance and foul bottom could set up stern shaft vibration. In this case the vibration was not so severe as only one shaft was involved.

This occurrence has been described in detail because of its bearing on subsequent events. While undergoing modernization, a leaky joint between two of the sleeve sections which cover the stern tube shaft and protect it from corrosion, led to the discovery of a circumferential crack in the shaft. This piece of shafting has been

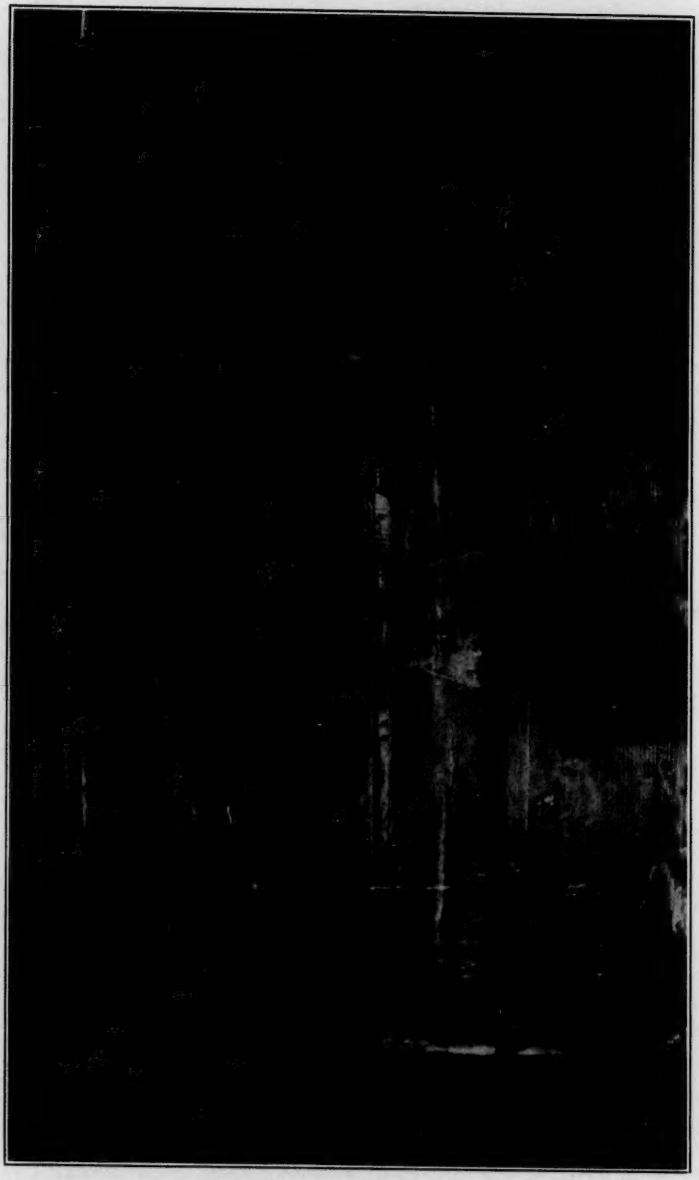

FIG. 1.—CORROSION—FATIGUE FAILURE OF A SHAFT.

examined at the U. S. Naval Engineering Experiment Station, in order to determine the cause of the damage to the shaft. The shaft was sectioned in such a way as to develop the extent of the crack. The outside of the shaft in two places showed definite corrosion. In one of these areas, the crack had penetrated the hollow shaft. The crack had progressed to some extent completely around the shaft and rupture was impending. Had the full power trials been held, the shaft undoubtedly would have carried away.

This is a typical example of a corrosion-fatigue failure. The location of the failed section was at such a point that it was at a node, in case any vibration stresses were imposed. This probably resulted first in the initial breaking down of the joint between the sleeve sections, which allowed access of a minute amount of salt water to the steel surface. The first stage of corrosion-fatigue, in which corrosion plays the more active part, then set in. Subsequently, when periods of high stress concentration existed in this area, incident to vibrations similar to those described above, there resulted a progressive type of failure, a portion of which is shown in Figure 1. From this incident another lesson is given the designer, namely, that locating joints and shaft sleeves at points of possible stress concentration is not desirable. In this particular case, corrosion-fatigue was the cause of failure, but the progress of the failure was also influenced by the vibrations described, coupled with a slight eccentricity of the bore of the shaft so that the shaft thickness in this plane varied almost as much as ½ of an inch.

In connection with long periods out of dock, on two occasions it became necessary to add a turn of packing in the stern tube stuffing box while the ship was in the water. This is a perfectly feasible operation if it is planned carefully and every precaution taken to prevent the contents of the stuffing box from being blown inwards by the sea pressure. With a split gland, little danger is present if only one-half of the gland is withdrawn at one time.

FIRE ROOMS.

It long has been recognized that the boiler room is the usual seat of many avoidable losses. In every vessel, it is probably quite safe to say that, regardless of her standing in the Engineering

Competition, some features involving waste can be found. The loss may be incident to either operation, boiler condition, condition of casings, blowers or other auxiliaries, or the tightness of the fire rooms. Boiler room operation is of extreme importance and unless frequent checks and analyses of results are made, unsuspected conditions will develop. Some of the conditions found in different ships and the steps taken to correct them will be enumerated.

Air tightness of boiler casings and fire rooms.—The task of keeping boilers and fire rooms air tight is neither as simple nor as easy as it sounds. And at the same time it is absolutely necessary for high efficiency. Generally speaking, looking for air leaks in a warm fire room or around a hot boiler is a job in which the enthusiasm of the searcher follows a rapidly descending curve as the search progresses. In this work a candle can be used to the best advantage, for it quickly and surely locates a leak. Visual inspection of many joints can be made with the boiler idle, and the smudge test is also of value.

One of the first steps in boiler room economy should be toward air tightness. In one case, it is remembered that every one of the boilers (B. & W. Marine type) was literally a sieve as regards tightness of casings, and much labor was necessary in order to correct this. It seemed that every joint was leaking in some degree. It was found also that the rope asbestos packing between the sinuous headers was most difficult to keep tight and the casing doors were therefore sealed. After using various mixtures for use as sealing compound and finding none satisfactory, a formula supplied by the Engineer Officer of one of the cruisers was tried and gave excellent results. The paste consisted of a mixture of four parts of flake asbestos and one part of dry powdered red lead (by volume), which was mixed into a paste by adding linseed oil. This mixture was used generously and to good effect on all the boiler casing joints. As the mixture is red, the boiler casings looked as though they had been camouflaged after all the leaks had been sealed.

Air leaks on the boilers were found in unsuspected places. At the joint between the steam drum and top horizontal casing, which is well covered by heat insulation, all the original asbestos packing

had deteriorated and disappeared. This allowed air to escape from the fire room into the uptake. As the joint clearance was fully ¼ inch for the length of the drum, a large loss of air occurred. Smudge tests were made, also, to show up additional leaks, and it was found that the horizontal brick baffles (roof baffles) were anything but tight, although the joints between bricks appeared so. To correct this, a very thin wash of fire clay was applied, effectively, over the top of the whole baffle. The conditions found are illustrative of what often happens in many machinery installations. The manufacturer builds a unit, taking the necessary steps to insure that it is as perfect as he can make it. Thereafter, many of the essential built-in features are either forgotten or unknown by the user and, consequently, unhealthy conditions develop and are not corrected. An originally tight boiler casing may soon become as bad as those described, because of warping, expansion and contraction, or vibration.

Air leakage of another type was also found to be considerable and worth correcting in this installation. The Babcock and Wilcox marine type boilers were arranged with three boilers in each fire-room. Each boiler had seven Cuyama burners. When not in use, it was impossible to prevent considerable air leakage around the burner registers, as the register doors only make metal to metal joints with the register framework. Consequently, when less than three boilers were in use in a fire-room, much air leakage could take place around the registers of the idle units. This was remedied by making canvas covers, which could be tied around the periphery of the register. These were easily applied and quickly removable. This produced large dividends in port, as only one boiler was required for auxiliary purposes, and this air leakage in the other two was thus eliminated.

Forced draft blowers.—The blowers on one ship were responsible for considerable air leakage. Each fire-room had three forced draft blowers. All the poorly fitting blower flaps were found warped and were renewed by the ship's force. Probing the blower casings with an air speed indicator (anemometer) showed large leakages inside every casing. On dismantling the casing to determine the cause, it was discovered that the upper or pivoted edge of the blower flap did not make a good joint with the blower casing in either the closed or open position. To correct this, a flex-

ible joint was provided by using a strip of wire inserted asbestos sheet, one edge being secured to the edge of the flap and the other to the casing. In this same case, blower rooms and air ducts were all searched with the air speed indicator to determine if any obstructions to air flow existed. Unfortunately, most of the obstructions, and there were many, were permanent, and correction was therefore impracticable. Numerous lockers in the air ducts, however, were given new locations. As blowers are most sensitive to interferences in free air flow, both on the intake and discharge sides, it is important that these be removed when possible. It was evident that if consideration had been given this point, many such obstructions on the suction side would have been avoided.

In this ship the results of the work on the blowers and boilers, solely in the line of preventing air leakages, was productive of perhaps the greatest single saving effect. By actual test under port operating conditions when burning oil at the average rate, the blower revolutions were reduced about one-half. The fuel saving in port was immediately apparent.

AUXILIARIES.

The maintenance of all auxiliary machinery in perfect condition has always been considered of maximum importance. Little can be done in this respect, outside of the Navy Yard, as far as the main propelling machinery is concerned, but practically all auxiliary units can be handled by the ship's force. Although inefficient units may not show up perceptibly on the underway performance, the "at anchor" results are visibly affected. Much can be done, by regular inspections and overhaul, to forestall the development of conditions which later may involve lengthy or major overhauls. As a rule, when a unit gives an indication of requiring attention in even one particular, it will be found profitable, then, to give the entire machine a careful inspection throughout, and overhaul as necessary. The word "overhaul" is too frequently used to describe cursory repairs. An overhaul of a piece of machinery that is not thorough in every sense of the word invariably means more repair work soon thereafter. It is well to remember, also, that if bad conditions are found in one unit, of which there are others installed, then the same conditions, invariably, either will be found,

or will be on the point of manifesting themselves in the other similar units. Remembering this often will save breakdowns or lengthy repairs.

HEAT INSULATION.

In most ships, both Naval and marine, there is great possibility of improvement in the matter of heat insulation alone. In the case of the *Idaho* much effective work was done by the ship's force. No steam and exhaust line flanges or valve bonnets were found to be covered. All of these represent most efficient black body radiating surfaces. The center engine room access ladder was directly over the port main steam line cross connection which carried all but two of the large control valves and main stop valves for the port engines. Entering and leaving the engine room meant going through a barrage of the heat waves emanating from these uncovered surfaces. It was anything but pleasant in the tropics and bad enough in the sunny climes of California.

This situation was perhaps instrumental in starting a definite campaign for insulating all such bare surfaces. The main steam line received priority, after which the whole auxiliary exhaust line, the auxiliary steam lines and all drain lines received attention in their definite order of importance. The insulation on drain lines beneath the floor plates was in bad condition generally and, in the case of the feed heater drains showed a drop of ninety degrees in temperature between the forward fire room and the after end of the engine room. It is believed that, where practicable, all drain lines should be run overhead. In the bilges they are out of sight and usually out of mind and it is most difficult to keep the insulation intact. Furthermore, they are inaccessible and as a rule much extra piping is required when run beneath the floor plates.

A definite plan for correcting these conditions was instituted. A study was made of the various materials available in the standard stock catalogue and small requisitions made for those thought most suitable. By a little experimenting, it was found that flanges could be neatly and expeditiously covered by using strips of a ¼-inch soft asbestos board which could be easily cut, bent, and shaped. In the case of flanges and fittings, two layers of the board were wrapped around and secured in place with wire. The usual wet

flake asbestos was used to fill in and build up around the form thus made and the whole was covered with standard "lagging" canvas. Valve bonnets were even easier, as a strip of the board was used to form a circular dam and the whole bonnet generously covered with the flake asbestos which then was covered with the canvas. In all cases the canvas was painted with a heat-resisting paint, which kept its white color and did not become faded and dingy looking.

The engine room division started the work, and four men under a competent petty officer were instructed and soon became as expert as the Navy Yard "laggers" and their work presented an equally good appearance. At least two of this force were kept at this almost continuously for more than a year. Other division details were taught by these men.

It is regretted that no record was made of the total number of square feet of radiating surface covered in this way. The results were gratifying in many ways. From the standpoint of appearance, the gradual disappearance of the unsightly black surfaces was most pleasing. The covering of the exposed surfaces of the auxiliary exhaust line was equivalent to supplying additional heat units for the evaporators and feed heaters. Under port conditions, when the exhaust steam available was hardly sufficient for these purposes, the extra heat units were much appreciated. The resulting increase in the port distilled water output was most welcome. The effect on the port fuel consumption was, of course, more apparent than on the underway figures.

Any argument offered against the covering of flanges and bonnets is not well taken. The general objection that it hides any leaks which develop does not apply, for a drip or blow is forced through the insulation almost immediately. The valve bonnets are covered in such a way that the glands can be tightened or repacked without disturbing the covering. Also, well covered joints are not as prone to develop leaks as bare surfaces, for the reason that more equable temperatures exist throughout. If the covering is applied as described, its removal and subsequent replacement is not difficult when repairs are to be effected. Moreover, it can be done quickly. The pudding-bag method of insulating valve bonnets, flanges, and fittings is recommended also.

The covering of bare hot metal surfaces has been tried and found feasible in the case of reciprocating steam pump cylinder heads, valve chests, and other exposed surfaces. Galvanized iron covers can be fabricated, which are easily removable. A layer of thick asbestos board is secured to the inside of these covers. This method of insulation adds both to the efficiency and to the appearance of the units.

MISCELLANEOUS IMPROVEMENTS.

Many changes of a minor nature are generally possible. Although the returns from any one may not be great, it is usually the combining of several such small savings that provides a total saving which is most gratifying.

Saving in make up feed.—On every ship a sizable decrease in the amount of make up feed used is possible. It is felt that in a marine plant more attention should be given to this item. The ideal to be strived for should somewhat approach that of the mercury boiler and turbine plant where liquid losses are *nil*. Ordinary leaks are inexcusable. Many unavoidable leaks can be greatly lessened.

The usual gland or stuffing box leakage can be prevented largely by trueing up all rods and valve stems and installing the correct packing properly. The use of the new semi-metallic grades of packing will be found beneficial. Stuffing box sizes should be checked and altered where not in accordance with the prescribed standard. On the main air pumps in one case, the boxes apparently were originally designed for metallic packing but soft packing was used. Thirteen rings of the three-quarter-inch size asbestos rod packing were required. To correct this a soft metal, split distance piece, was installed so that six turns were ample. Also, the asbestos type rod packing was discarded and semi-metallic installed. This packing had five or six times the life of the other style on these pumps. The plastic metallic type of packing now furnished will be found invaluable for use on small valve rods and also on small rotary pumps. Best results will be obtained with these types of packing only if the rods are true and smooth.

In the *Idaho* installation all of the main turbine glands exhausted to the atmosphere. A small exhauster condenser was built aboard

and installed on one of the six turbines and when the saving was determined, similar units were put on the remaining five. The daily saving of twelve to fifteen hundred gallons was equivalent to more than one hour's operation of the evaporators at full capacity. In this same connection, the glands of many of the auxiliary turbines were the old grease packed design which was anything but effective. The Navy Yard was persuaded to install carbon packed glands on the circulator turbines during the Yard overhaul. The repair ship cast gland bodies for other units and the ship made the necessary alterations and installations. A sizable total saving in make-up feed requirements resulted from this item alone.

Thermostatic valves.—This type of drain valve can now be purchased for all the galley steam kettles and urns. They undoubtedly reduce the galley steam consumption and do away with the usual frantic calls from the galley for more steam and requests to by-pass the galley steam traps. The valves give no trouble, operate perfectly, and because of this, it will be possible generally to reduce the steam pressure in the galley line considerably. They are also of value for laundry, press shop, and similar equipment.

Aids to operation.—A certain number of steam gages and thermometers are necessary for the operation of any plant. The installation of additional numbers of these often can result in better and more efficient operation. The location of these instruments is also of importance. They must be where the man responsible for the operation of that particular unit or units can see them easily while manipulating the necessary valves. For instance, a vacuum gage on the main instrument board is of little value to the man on watch at the condenser auxiliaries. Many cases can also be found where no provision has been made for a gage. In such cases, provision should be made. It was surprising, in one instance, how the economical regulation of the main circulators was improved by providing a steam gage on the circulator turbine steam chest and locating it where the operator naturally faced it when working the throttle valve.

One of the most valuable aids for use in checking condenser and evaporator operation was found to be a small absolute pressure gage. This was made up aboard, but it is not easily done, for it

requires, in addition to a good piece of glass tubing, much patience. One end of the tube is sealed in a blow torch flame. It is important that the end of the small, almost invisible pipe, which may form when sealing the end, be covered over with an extra layer of glass to insure that a perfect seal is made. The tube is then bent into a U shape, with the open end brought out horizontal from the U for attaching a rubber tube as a connection. If the glass is good and the operation is not hurried, this part is easy. The length of the U determines the minimum reading obtainable. For instance, a six-inch leg means that readings from zero to 6 inches of mercury (0-3 pounds) in absolute pressure, or 24 to 30 inches in vacuum can be made. No corrections for barometer are necessary. The difference in the length of the two mercury columns gives the reading direct as absolute pressure in inches of mercury. The patience factor comes in when getting the tube filled (the closed leg and about a half-inch of the open leg) with mercury. The glass must be clean and dry. Washing first with alcohol, followed by a rinsing with acetone or ether, gives good results. It is then advisable to put the tube over a small, hot plate to get it thoroughly dry. Meanwhile, clean mercury must be heated moderately to drive off any air bubbles. Introduce the mercury in small quantities with a medicine dropper and then commences a puzzle as intriguing as any you ever encountered. Before you succeed in getting the mercury safely and tightly into the closed leg with no air space in the closed end, your interest in the puzzle may change—you may find need for some golf course phraseology that you use subsequent to a missed putt or a topped drive. The easiest way to work the mercury into the closed leg will be found to connect the gage to some source of vacuum, as a condenser, and then, by trial and error, rolling the gage on its side, gently tapping it, tilting it, or laying it on its side, you can eventually work out the last trace of air and get the closed leg full of mercury with no air space above it. Care must be taken not to allow mercury to get into the condenser. The gage should be checked against a mercury column in the Navy Yard or on a repair ship. A rough check can be made against a vacuum gage known to be accurate, but too much reliance should not be placed on it. If two gages are made and they check against each other, you can be

reasonably certain they are correct. Do not expect to have the first gage completed without a hitch. One or two tubes at least should be broken in the process, unless you are expert at bending glass and deft in filling the tube with mercury. The gage should be mounted on a small wooden board about 4 inches by 7 inches, on which has been glued a piece of cross-section paper (ten divisions per inch) to serve as a scale. Once done, the portable gage gives a fund of accurate and valuable information. It was interesting in the case of the *Idaho* to find that the pressure drop across the tube bank in the main condensers was as much as 1.8 inches at full power, which gave a satisfactory explanation of the low vacuum at full power. Moreover, such a gage obviates ever taking a vacuum or compound gage to the Navy Yard instrument room for testing, as it can be done right aboard and the gage set or adjusted quickly and easily. By providing additional outlets at every such gage, it is only an hour's work to check them all in place. Another advantage of the absolute pressure gage is that it can be read accurately to tenths of an inch and hundredths can be accurately estimated. The home-made type is inexpensive, compact, and reliable.

Methods of operation.—Time spent in study of operating methods and actual experimenting with different conditions is time well spent. Actually making special performance runs on different auxiliaries and constructing the characteristic speed, steam pressure, water rate, and output curves provides valuable information and also instruction and testing experience for engineering officers and petty officers. Once constructed, subsequent operation can be checked readily and any falling off in efficiency noted. Construction of these curves from actual runs aboard ship is desirable, even though curves have been furnished by the manufacturers.

In conducting full power trials, a set procedure should be fixed, and if the plan is correct and the personnel instructed, no trouble should be experienced in meeting the prescribed maintenance of constant conditions, even though the required full speed may not, for the time being, be possible to meet. In other words, boiler power and steam consumption should, at all times, be so balanced that there is at least a slight excess in the favor of the boilers.

Exceeding the boiler output for even a short time, in the effort to make the desired R.P.M., is foolhardy.

The trial procedure used on the *Idaho* gave excellent results for three successive years. Balanced conditions were maintained closely at all times and the throttles entered into the picture but little. The boilers were used almost entirely for speed control, first by increasing the oil pressure in the initial stages, with all but two of the sixty-three available burners in use, until full power speed conditions were well satisfied; and subsequent to this, by manipulation of the two available burners. Once the throttles had been "inched" up to the required speed, their position was never changed. In two of the three fire-rooms (three boilers per fire-room), it was then possible, with this plan, to set all conditions for maximum boiler efficiency. In the control fire-room, as not more than two additional burners were ever required, only slight changes in blower speed were required.

With the above system of control, the required speed was reached easily. The throttlemen had a leisurely time of it figuring revolutions, as they were forbidden to touch the throttles except when directed. The fire-rooms were hardly aware that anything out of the ordinary was going on, as there was nothing to do but keep burners cleaned and maintain constant conditions. Feed check valves were not allowed to be changed more than a tenth of a turn of the valve wheel and hence, no boiler ever was guilty of "hogging" the feed, and constant feed pressure was consequently easy to provide.

During the first run made, in the middle of the maximum speed period, a variation in the method of control was hit upon as a result of watching conditions and keeping in close touch with every change which was taking place. Since the attempt to establish steady and stable conditions had been realized, it was noticed that the starting of a bilge pump or any other small unit was sufficient to upset the balance and to require some small change to be made. As the engine room bilge pump served many spaces and was much used it was decided to use this pump as a control unit. Consequently, when the steam pressure next reached the upper limit of the prescribed pressure range, this pump was started at moderately high speed without making the usual reduction of one burner.

The boiler pressure rose an additional four pounds, almost to the safety valve setting, finally stopped, and then started downward at a gradually increasing rate. On reaching the middle of the pressure range, as it was realized then, that there would be some lag between cause and effect, the pump was stopped. As expected, the downward trend of the steam pressure continued, slowly decreasing in speed. Finally the black hand stopped and hung in a balance for about thirty seconds, which seemed interminable to those worrying whether it would ever start upward, which eventually it did. At the middle of the range the pump was started and the cycles repeated. The procedure was refined gradually by juggling the bilge pump setting so that the time required for each cycle was lengthened greatly. This made it possible also to keep the control fireroom under steady conditions except when routine additional power demands made changes necessary. The scheme was used about two hours of the full power trial and for the entire four hour smoke prevention run, and on all power trials thereafter.

This incident has been narrated principally to illustrate how closely a marine power plant can be adjusted and maintained on steady conditions and how easily and finely these conditions can be controlled. To the operating personnel it was a convincing demonstration of the ideal to be attained under any condition of operation.

Daily Operating Results.—The operating results obtained from day to day should be consistent and any variation therefrom investigated. It will be found that if accurate data involving the principal variables which affect the oil consumption, are collected, analyzed, and averaged, a valuable schedule can be made up of what the hourly requirements should be under normal conditions. By making up a form covering a week's operation (for port conditions), with the determined hourly requirements shown at the left-hand edge for each hour of the day, and entering the results being obtained from hour to hour, a quick and ready means of determining the existence of any unusual conditions is provided. After using this plan for a few weeks, it will be found generally that the previously determined hourly fuel requirements can be revised downward. The keeping of this hourly information sheet will be found of great value in many ways.

GENERAL.

Some general observations, principles, practices and maxims, which have been found of value are included under this heading.

In cleaning most heat transfer apparatus the use of a pump for circulating the hot cleaning solution is a time saver and adds to the thoroughness of the work. The violent agitation and circulation produced by the pump produces a scrubbing and cleaning action which is much more effective than any which may result from the natural circulation induced by boiling out.

The degree of maintenance of clean water sides of boilers is largely a function of the attention paid to the condition of the water in the boilers. The casual or indifferent addition of a pound or two of compound is in no way sufficient. Much time, labor and B.T.Us.—and an occasional tube failure—can be saved by careful attention to this important feature of boiler operation. The basic principles of internal boiler feed water conditioning as applied to Naval practice can be found in the Bureau Manual and Bulletins of Engineering Information. The writer is convinced that carrying out the recommendations in the above as prescribed for a complete 700-hour steaming interval will do far more toward removing any scale present than several boiling-out periods.

As regards determining boiler cleanliness the usual method of boresighting the tubes, using a light or spot light from either end, is of little value. Many scales are most deceptive and look exactly like the tube metal, especially after being brushed. Even under the ideal conditions of short, straight tubes this method of examination generally gives erroneous impressions. For this reason the writer has made a practice of always removing the worst appearing tube or tubes from a boiler at each overhaul. Splitting such tubes makes accurate examination possible and gives a good indication of conditions existing throughout the boiler. The removal of such a sample tube at each overhaul is recommended for general practice.

The old maxim, "cleanliness is next to godliness," might well have been coined expressly for the operating Engineer. Practically every machine depends on cleanliness for its efficient and proper functioning. Clean boilers, clean turbines, clean oil and

bearings—all are necessities. Many a nut, bolt or tool adrift in a steam line or turbine casing have been the cause of much damage; a coin in a boiler tube is on record as having caused one boiler casualty; in the case of electrical machinery, cleanliness is even more important. While on the shake-down cruise in the electric drive battleship some steel chips in the air duct of one main drive motor caused damaged coils and much fatiguing repair work. Eventually, in a different main drive motor, a steel chip which had found its way into the air duct, either before or after the motor was installed in the ship, caused a short circuit between phases and a fire in the insulation. This fire resulted in damage to all the stator and rotor coils, as well as to some of the laminated iron. As a result, some of the iron had to be relaid and the entire stator and rotor rewound. Not only was the labor and expense involved great, but the placing of the ship in an operative status was delayed about two months. All of the above illustrations are a few outstanding examples of what occurs daily for lack of cleanliness.

Another maxim found useful and applicable, especially to engineering, is "Eternal vigilance is the price of safety." Many a burned-out bearing, many salted-up boilers, many burned-up boilers, innumerable accidents involving personnel, as well as machinery damage, and casualties of most every description of daily occurrence can be ascribed to carelessness and lack of vigilance.

Engineering duty invariably is filled with interesting and, at times, harrowing experiences. Situations are bound to arise which call for serious thinking and planning through many worrisome hours. But most engineering plants are so arranged, and the tools and appliances available are so diverse, that there is generally some way out of every difficulty. All in all, a Naval machinery installation may be looked upon as a large testing and experimental laboratory. Opportunities abound for studies of machinery performance and methods of operation, constructional and design features, as well as means and methods of upkeep and maintenance.

As a general rule, results obtained in Engineering do not come easily or quickly. Regardless of whether the ship is relatively poor, both as to machinery condition and operating efficiency, or an efficient ship in every way, any results obtained are slow in

manifesting themselves, regardless of the intensity of the efforts being made for improvement. Machinery condition, operating methods, and the general ability and knowledge of the personnel, are all factors in every-day performance, and changes affecting any one of them necessarily take time, patience, and perseverance. He who goes to a ship which has a relatively poor standing and expects by swift-moving methods to locate and correct, in a few short months, the sizable deficiencies thought to exist, is quite likely to be disappointed.

It is believed, also, that in every ship, regardless of her standing in the Engineering Competition, there is possibility of improvement. An installation at the peak of its efficiency is daily undergoing adverse changes tending to lower that state. Even though the personnel be highly competent in operating procedure, a change in personnel involving only a few men or perhaps an officer, may have an effect in the wrong direction, which, unless guarded against, will persist until the responsibility for it is located.

This leads up to a striking difference between engineering aboard ship and the other departments. Engineering is an every-day job that must go on at an even, steady pace day after day throughout the year. There can be no let-up, and with the exception of the yearly full-power run there are no peaks throughout the year as is the case with gunnery. This condition, however, is conducive to stabilizing the interest of the personnel, for each day's performance is something to strive for and to do a little better than it was done the previous day.

Assignment to a vessel low in the engineering competition does not necessarily mean that there exists great or unusual possibilities for improvement. Nor should assignment to one of the leaders in the competition be a deterrent to aggressive action for further improvement in her standing. The possibilities are present in both cases.

The Engineering Department aboard ship is very much on its own and the tendency in many cases is to let the engineers shift for themselves. This condition certainly is a desirable one and can be fostered in many ways. For instance, the line of endeavor in which the engineers are most conspicuous is in the operation of the ship's power boats. In this work the engineers can make their

presence felt like a sore thumb by having frequent boat breakdowns and thereby bringing much undesirable attention to themselves. On the other hand, smooth-running boats will give just the opposite effect. Careful attention given to the employment of high-class boat engineers and repair men and maintenance of these important auxiliaries always in first-class running order is well worth while. In many other ways the Engineer Department must give service to other activities. Being one step ahead of the game in these matters calls for a little foresight and thought and often prevents an unpleasant situation.

Many of the ships high in the engineering competition are accused often of being both dark ships and dry ships. Such a reputation is both unsavory and unnecessary. Adequate lighting and proper supply and distribution of water are necessary for health, comfort, and morale. The electric load can be reduced in other ways and the normal supply of water required based on man–gallons per day should always be supplied. Water troubles are invariably caused by improper distribution. Education and an occasional resort to rationing for a period of a week works wonders, and thereafter a free spigot is respected.

In conclusion it is desired to refer to the graphs representing the performance of one vessel. The improvements shown were effected over a period of three years. The graphs depicting port performance show the average hourly fuel expenditure for each month. These points have been connected together, although the connecting lines have no significance other than to show the month to month consistency of operation. In addition to the monthly averages shown by the graphs, the yearly average hourly port fuel consumptions are shown to one side. The two years following commissioning are not shown for the reason that they were excessive and therefore were not considered representative of actual conditions. Curves "A" and "B" showing underway results, were constructed under identical conditions one year apart, when advantage was taken on trips to or from the Navy Yard to make test runs at speeds from 8 to 15 knots to obtain data for making up these curves. The runs at each speed were five hours in duration, the first being used for steadying down and the last four hours averaged to obtain the points for the curves. Surprisingly

smooth curves, which required practically no fairing, were obtained. These curves were of considerable value in checking operation whenever steady run conditions existed. The data tabulated for yearly underway performance should be interpreted after considering that during the engineering performance years 1928-1929 and 1929-1930, the vessel remained out of dock for full 12-month periods, whereas in the year 1930-1931 docking occurred after

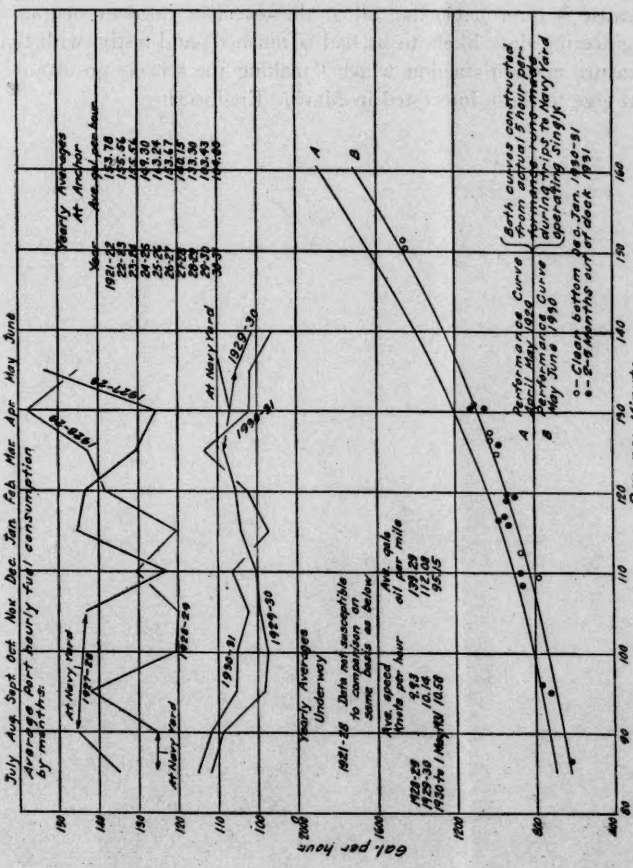
July Aug. Sept Oct Nov Dec. Jan Feb Mar Apr May June
Average Port hourly fuel consumption by months.
At Navy Yard 1927-28
At Navy Yard 1926-27
At Navy Yard 1926-28
Yearly Averages At Anchor
Year Ave. bbl. per hour
1921-22 133.70
22-23 135.66
23-24 149.30
24-25 143.24
25-26 133.67
26-27 128.15
27-28 133.30
28-29 145.93
29-30 140.00
At Navy Yard 1929-30
1929-30
1928-29
1930-31
Yearly Averages Underway
1921-28 Data not susceptible to comparison on same basis as below
Ave. speed Knots per hour
Port gals. oil per mile
1928-29 8.13 130.29
1929-30 10.14 112.02
1930-31 Navy Yd 10.50 95.15
A Performance Curve April May 1928
B Performance Curve May June 1930
Both curves constructed from actual 5 hour performance runs made during trips to Navy Yard operating singly
o - Clean bottom Dec. Jan. 1930-31
• - 2-5 months out of dock 1931
Bbl. per hour
Gal. per hour
Rev. per Minute.

FIG. 2.—UNDERWAY AND AT ANCHOR FUEL CONSUMPTION.

eight months. Some of the apparent improvement shown during the last year must be ascribed to this difference in docking intervals.

The writer looks back on every tour of engineering duty with mingled feelings of regret and satisfaction. With regret, because belated reflection invariably calls forth some opportunity missed, some interesting experiment that should have been tried, or some line of effort and attack which was not prosecuted effectively to completion in time to give the desired results; with regret, also, because it is probable that all of the direct or intimate operating engineering duty likely to be had is finished; and lastly, with that pleasure and satisfaction which "making the wheels go around" can give to those interested in Marine Engineering.

ALTERNATING CURRENT IN THE ELECTRIC PLANT ON NAVAL VESSELS.

By T. E. Cassey, Member.*

In specifying alternating current for the complete electric plant installation for the new 1932 Destroyers the Navy Department has entered into a new field of marine electrical design. It may not be definitely predicted what the results of this departure from the conventional direct current installation will be, but at least this development will lead to the solution of the many interesting technical problems involved. It is the purpose of this article to discuss the numerous problems which confronted the engineers in the development of the alternating current electric plant for these naval vessels.

Alternating current applications are not new on naval vessels, having been in use for a number of years for interior communication and fire control purposes and for the universally known electric drive on battleships and airplane carriers. Alternating current has also had other minor uses on shipboard.

When the 10,000 ton class of cruisers was being projected, consideration was given to the use of alternating current for the electric plant but investigation of the problem, as then presented, did not lead to the adoption of such a system for these vessels. Only with the authorization of the 1932 Destroyers was the adoption of the complete alternating current electric plant to be realized.

The advantages of alternating current generation and distribution for land installations are so well recognized that no discussion is needed of the relative merits of the alternating current and direct current installations for many industrial uses.

For the naval vessel a different situation exists. Design difficulties, always encountered in the electric plant development for the fighting vessel, may be expected to increase with the use of alternating current. The alternating current ship will have certain

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distinct advantages and some disadvantages in comparison with the direct current ship. Briefly, the principal advantages gained by the use of the alternating current electric plant lie in the simplicity of design, increased ruggedness and greater dependability of A.C. generators and motors, in the reduced requirements for motor control equipment, in the decrease in motor and generator spare parts, in the reduction in the number of motor generators, in economy in the maintenance of the plant, in increased efficiency in operation, in the elimination of sparking commutators, in the absence of electrolytic action from stray currents, and in the reduction of magnetic disturbing effects on the magnetic compass.

The principal difficulty in the alternating current system for the naval vessel lies in the establishing of the stand-by or auxiliary power supply for vital auxiliaries, now provided by storage batteries. With alternating current there will also be added complication in switchboard designs, a more involved distribution system, and the need of hydraulic variable speed gears or other means for obtaining necessary speed control for certain auxiliaries.

In initiating the design of the alternating current electric plant for the 1932 Destroyers, the designers were first confronted with the questions of voltage and the extent to which the application of alternating current could be carried on shipboard. The voltage, after careful consideration, was finally settled at 230, to permit the use of the commercial standard of 220 volts for motors. Consideration was given to the use of 440 volts but this voltage was abandoned for destroyers in view of the probable danger to personnel and because of the relative small power requirements for this type of vessel.

On the direct current ship it is found necessary to provide an alternating current source of power for applications not suited to direct current. Likewise on the alternating current ship a direct current source of power must be provided for applications that do not lend themselves to operation on alternating current. The apportioning of the use of alternating and direct current power is one of the vital questions affecting the design of the alternating current ship, and the solution of this problem will not be the same for the different types of vessels.

In writing the specifications for the new destroyers, based on a summary of the power requirements for the vessels, a generating

plant of two 125 KVA turbine driven alternators, 230 volt, 3-phase, 60 cycles, was decided upon. Exciters were specified as direct connected, 120 volt. For the establishment of a 120 volt bus for lighting service, 230/120 volt transformers were provided. For the D.C. power demands, primarily for searchlights and storage battery charging, two A.C. motor driven 120 volt D.C. generators were specified.

With a generating plant of two turbine driven alternators and two motor generators thus fixed, investigation was started to determine the characteristics of the proposed alternators. At the time of this investigation, no definite information was obtainable as to the voltage regulation possible for small alternators of the characteristics selected, and also as to the performance of two such machines operating on a bus and subject to the sudden fluctuations of load experienced on naval vessels. In the absence of definite information on this point there was written into the specifications for the vessels the requirement that the voltage regulation of these machines should not exceed 12 per cent (rise) when the load on the generator is reduced from 100 per cent to 0 load in one step, and should not exceed 20 per cent (drop) when 100 per cent load is applied to the machine in one step. It was further provided that voltage regulators should be installed which would automatically control the A.C. voltage within 2 volts plus and minus under the above named conditions. The values specified for voltage regulation for these machines were more or less arbitrary, and were indicative of the surprising lack of definite information as to the performance of two small alternators working on a bus. The initial schemes for controlling these two machines were developed on the basis of independent operation of the machines on separate buses, with double throw switches on all feeders. This was changed, however, to require synchronous operation of the machines with a view to forcing this issue to a satisfactory development.

Since the issue of the 1932 Destroyer specifications, information has been obtained justifying the expectation of good voltage regulation with small machines as proposed for installation on these vessels. Thus for future A.C. generating plants for vessels, it will be possible to specify for synchronous generators

operating on a bus, normal voltage regulation within plus or minus 1 per cent and a drop in voltage not to exceed 5 per cent under a sudden load of 50 per cent of rating, the time of return to normal voltage in this case not to exceed 2 seconds. With such characteristics as these, the successful operation of the A.C. turbo-generator plant for naval vessels appears to be assured.

It may be of interest to describe briefly the type of voltage regulator which it is proposed to provide for the 1932 Destroyers to meet the voltage requirements given in the preceding paragraph. This type of voltage regulator combines in one device the ability to alter the machine excitation at a rate and in an amount which is proportional to the voltage variation to be corrected. This equipment, in addition to the voltage regulator proper, consists of a motor operated rheostat and a series of high speed relays controlling a field changing resistance. Small changes in voltage are controlled by the motor operated rheostat direct. For heavy disturbances, the relays operate to insert the field changing resistance in series with the rheostat, maintaining the voltage within the specified limits until the motor operated rheostat has moved to a position corresponding to the required excitation. As long as no change in excitation is required the device remains dormant. Other types of voltage regulators are available and it is possible that a more simplified equipment may be forthcoming for later vessels.

With the design problems of the A.C. end of the turbo generators no longer in question, the exciters and the source of D.C. power were taken up. Information first obtained from commercial sources indicated that the exciters should be separate machines, and accordingly the requirement was developed for A.C./D.C. motor generators to supply the necessary D.C. power.

This setup, however, was complicated by the requirement for some source of uninterrupted D.C. power for limited use in certain battle service circuits which would not entail the continuous running of large A.C./D.C. motor generators. It was then determined to increase the size of the exciters sufficiently to permit the supply of these small power demands, and accordingly a 50 per cent increase of exciter capacity was written into the specifications. Separate power circuits were to be run from the two exciters for this service.

After the award of contracts for building Destroyers No. 348, *Farragut,* and No. 349, *Dewey,* the contractor for one of the vessels proposed, as a means of saving weight, that the two A.C./D.C. motor generators be eliminated and the exciters be increased in capacity from 3Kw. necessary for excitation, to 25Kw. to provide for the supply of D.C. power from these machines, and that these machines be operated in parallel on a common bus. As this scheme was in line with the original ideas regarding the development of D.C. power supply for the alternating current ship, this proposal was adopted. In addition, as a means of further weight saving, the contractor offered a design of A.C. generator eliminating the necessity for transformers for the 120 volt lighting system by tapping the armature winding to establish the 3-phase, 115 volt lighting bus. This proposal has been tentatively accepted for machines of destroyer size.

Figure 1 shows a diagram of connections for the two 125 KVA machines and bus connections as originally developed for 1932 Destroyers. Figure 2 shows a diagram of connections for the system as developed to date.

Before leaving the subject of the A.C. generating plant it may be of interest to summarize the development made for this feature of the electric plant in the short time that the subject has been under consideration, this development of course being tentative and subject to revision as the building of the first A.C. ships progresses. This development may be given as follows:

A.C. generator characteristics determined.

Obtainable voltage regulation specified.

Performance of voltage regulators established.

Use of exciters for D.C. power supply in addition to excitation accepted.

Installation of taps on the A.C. armature windings for establishing 120 volt lighting bus accepted.

The distribution system for the A.C. ship differs in detail from the D.C. ship and is somewhat more complex, the complication being more marked the larger the vessel. For the 1932 Destroyers the distribution system is inherently more extensive than that of present destroyers, owing to the increased size of the vessels, and, as proposed, consists of a combined generator and distribution

switchboard, an auxiliary power or battery charging switchboard, an interior communication switchboard and fire control switchboards with the necessary interconnecting bus feeders and distribution feeders for the lighting, power, interior communication and fire control systems.

Generator and distribution switchboards for A.C. ships will in general be of larger size and somewhat more complicated than these switchboards installed on D.C. ships, in view of the additional control required for A.C. machines, the installation of voltage regulator equipment, exciter control, and the use of 3-pole, 3-phase switches and circuit breakers. The design of D.C. switchboards for naval vessels is standardized to a degree which permits, for a vessel of any given class and characteristics, the Naval designer to determine accurately, in advance, the space requirements, the weights, and the arrangement of the switchboards required for the vessel. The development of switchboards for A.C. ships, by reason of the inherent increase in size and the use also of unfamiliar switchboard equipment, will require careful study in order that these boards may not overstep the space and weight limitations.

The 1932 Destroyer generator and distribution switchboard, it is expected, will be approximately 16 feet long by 6 feet 6 inches in height as compared with 4 feet 6 inches long by 5 feet 8 inches high (exclusive of I. C. panel) for older destroyers. The board is not yet completely designed, but in Figure 3 there is presented a tentative diagram of connections for it which is of interest, as illustrative of the Navy's first A.C. ship main switchboard.

Some of the increased size of these switchboards is due to the normal increase in the electric plant for the larger vessels, but from the diagram shown in Figure 3 it is evident that the increase in switching for the alternating current plant requires the additional panel space involved. It will be noted that the diagram, Figure 3, shows the generator connections, as described in a previous paragraph, providing for the 120 volt 3-phase lighting bus being tapped off the armature windings. The diagram also shows the 120 volt D.C. exciter bus used as a supply bus for searchlights and battery charging as previously referred to.

The interior communication switchboard for the A.C. ship shows a distinct change from existing D.C. ship practice. On older

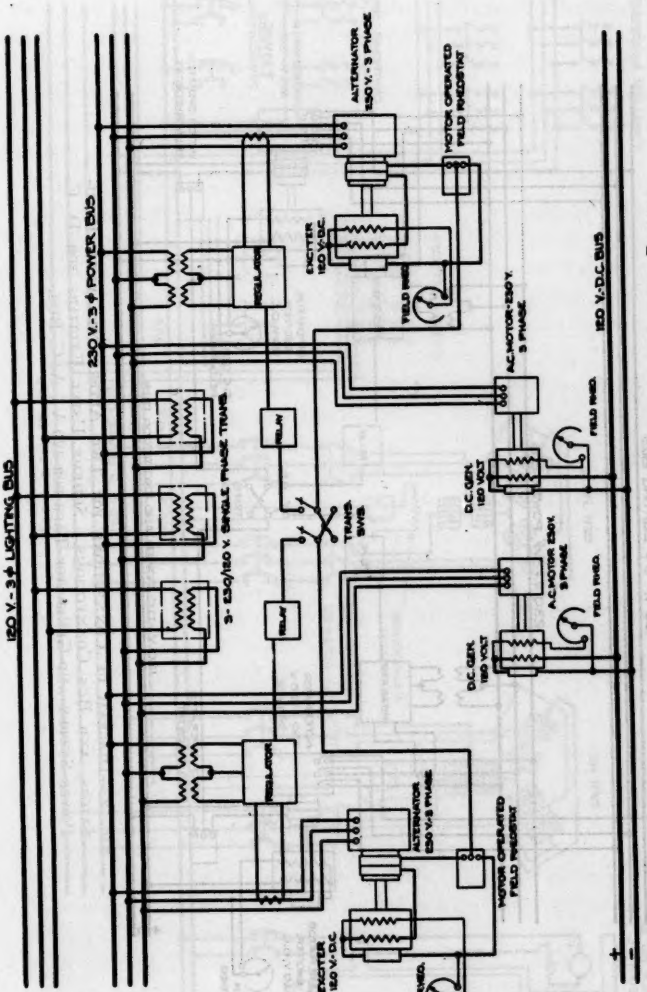


FIG. 1.—DIAGRAM OF CONNECTIONS FOR TWO ALTERNATORS WITH REGULATORS, AND BUS CONNECTIONS. SCHEME USING MOTOR GENERATORS FOR D. C. POWER SUPPLY AND TRANSFORMERS FOR 120 V., A. C. BUS.

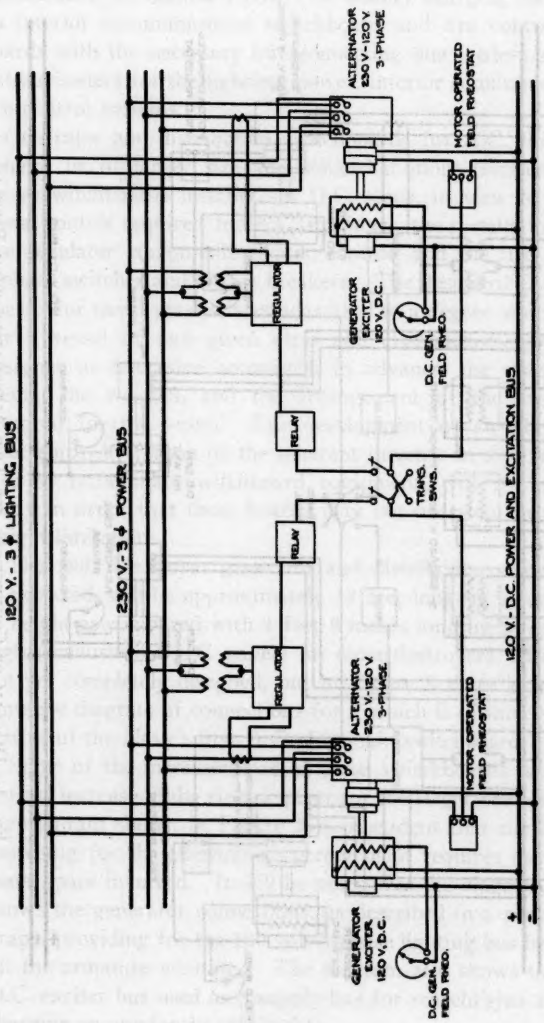

120 V. - 3 ϕ LIGHTING BUS
230 V. - 3 ϕ POWER BUS
ALTERNATOR 230 V - 120 V 3-PHASE
MOTOR OPERATED FIELD RHEOSTAT
REGULATOR
GENERATOR EXCITER 120 V, D.C.
D.C. GEN. FIELD RHEO.
RELAY
TRANS. SWS.
120 V. - D.C. POWER AND EXCITATION BUS
+
-

FIG. 2.—DIAGRAM OF CONNECTIONS FOR TWO ALTERNATORS WITH REGULATORS, AND BUS CONNECTIONS. SCHEME USING EXCITERS FOR D. C. POWER SUPPLY AND GENERATOR TAPS FOR 120 V., A. C. BUS.

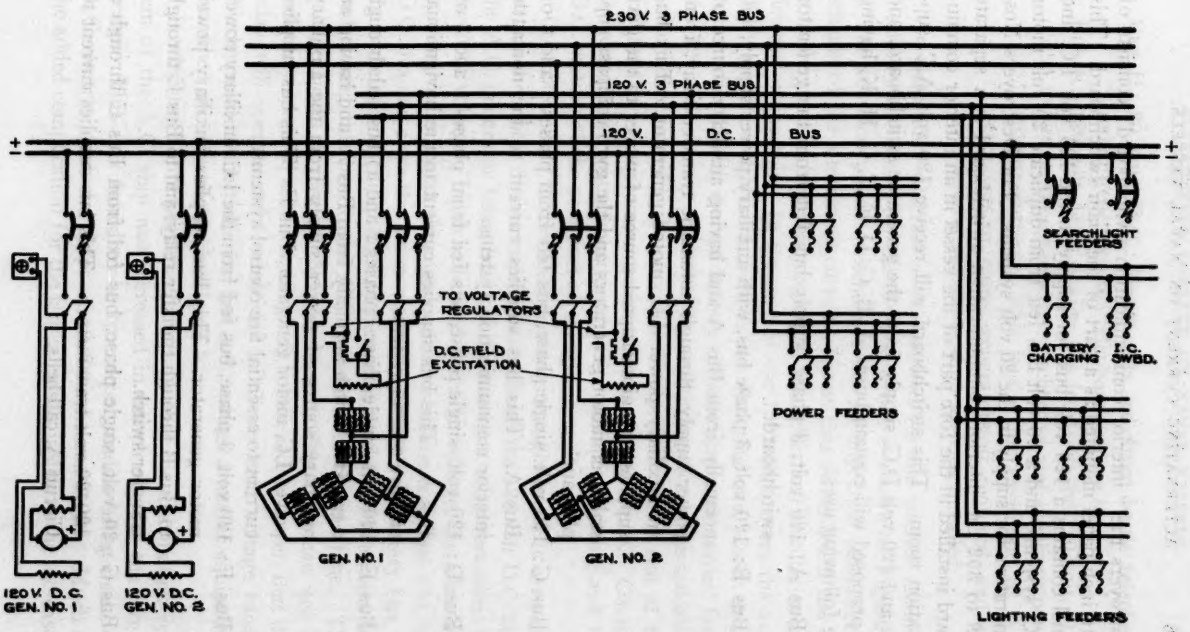


FIG. 3.—DIAGRAM OF CONNECTIONS FOR GENERATOR AND DISTRIBUTION SWITCHBOARD: TWO 125 KVA. MACHINES.

destroyers the "interior communication switchboard" consists of a 16 inch panel mounted as a part of the main switchboard. This panel contains a 125 volt bus for supply to the 125 volt I.C. and F.C. systems, and a 20 volt bus fed from duplicate 20 volt motor generators for supply to the 20 volt systems. For Destroyers Nos. 348 to 352 the interior communication switchboard is a separate board installed in the fore part of the vessel in an interior communication room. This switchboard will receive 120 volt A.C. supply and 120 volt D.C. supply from the generator switchboard, and as proposed will contain buses A, B, C, D, E, F, G, H, K, having the following uses:

Bus A: 120 volt, 3 phase, supply bus fed from the generator switchboard.

Bus B: 120 volt, 3 phase bus, with auxiliary power supply, fed normally from Bus A and having auxiliary source of power supply, through automatic transfer relay, from an auxiliary power I.C. motor generator. This bus supplies an uninterrupted source of power to the gyro stabilizer motor generators and the gyro compass supply panel.

Bus C: 120 volt, single phase, bus fed from phases 1 and 2 of Bus A. This bus supplies current to the nonbattle interior communication systems.

Bus D: 120 volt, single phase, bus fed from phases 1 and 3 of Bus A. This bus supplies current to the navigational and fire control systems.

Bus E: 120 volt, single phase (phases 2 and 3) bus fed through a transfer switch normally from Bus A, and having an auxiliary source of power supply from the auxiliary power I.C. motor generator, Bus F. This bus supplies current to essential fire control systems.

Bus F: 120 volt, 3 phase, bus fed from the I.C. auxiliary power motor generator. This bus supplies auxiliary power to Bus B through transfer relay, and to Bus E through transfer switch.

Bus G: 20 volt, single phase, bus fed from Bus C through a 120/20 volt transformer. This bus supplies current to Circuit A, call bells.

Bus H: 20 volt, D.C., 2-wire, bus fed from battle telephone motor generator and telephone storage battery.

Bus K: 120 volt, D.C., 2-wire, bus fed from the 120 volt, D.C. bus on the generator switchboard and auxiliary power storage battery. This bus is for excitation of A.C. generators of motor generators, if required, and any D.C. interior communication systems installed.

An elementary diagram of connections for the interior communication switchboard for the A.C. destroyer is shown on Figure 4.

The auxiliary power switchboard has an important place in the scheme of the alternating current ship as this switchboard ties up with the supply of auxiliary power to vital electrical auxiliaries and devices on the ship. In the case of the 1932 Destroyers the auxiliary power switchboard controls the charging and power distribution of a 120 volt storage battery. Two circuits are carried from the 120 volt storage battery bus to the interior communication switchboard, one for the operation of the D.C. motor end of the I.C. auxiliary power motor generator, and one for Bus K. Circuits are also taken from this battery bus for auxiliary lighting and for D.C. motors of the port fuel oil pumps.

Control of the I.C. auxiliary power motor generator is arranged so that the motor of this machine will idle on the 120 volt D.C. bus, the A.C. generator being held off the line by a relay controlled by the A.C. voltage. Upon failure of the A.C. voltage, the relay will operate to disconnect the motor of the machine from the 120 volt D.C. bus and close the motor circuit to the 120 volt battery bus, at the same time opening the A.C. circuit to the I.C. switchboard bus and closing this bus to the A.C. generator of the machine. The voltage limits governing the operation of this relay can not be definitely specified at this writing. It is believed, however, that this relay will operate on a sensitivity of 5 per cent voltage range. A scheme of connections for the A.C. voltage transfer equipment described above is shown in Figure 5.

It was stated in a previous paragraph that the distribution system of the A.C. ship was increased in complexity for the larger vessel. This was found to be true from studies being made of A.C. electric plants for ships of 10,000 tons and above. Much of the added complication for the larger vessels is due to the increased

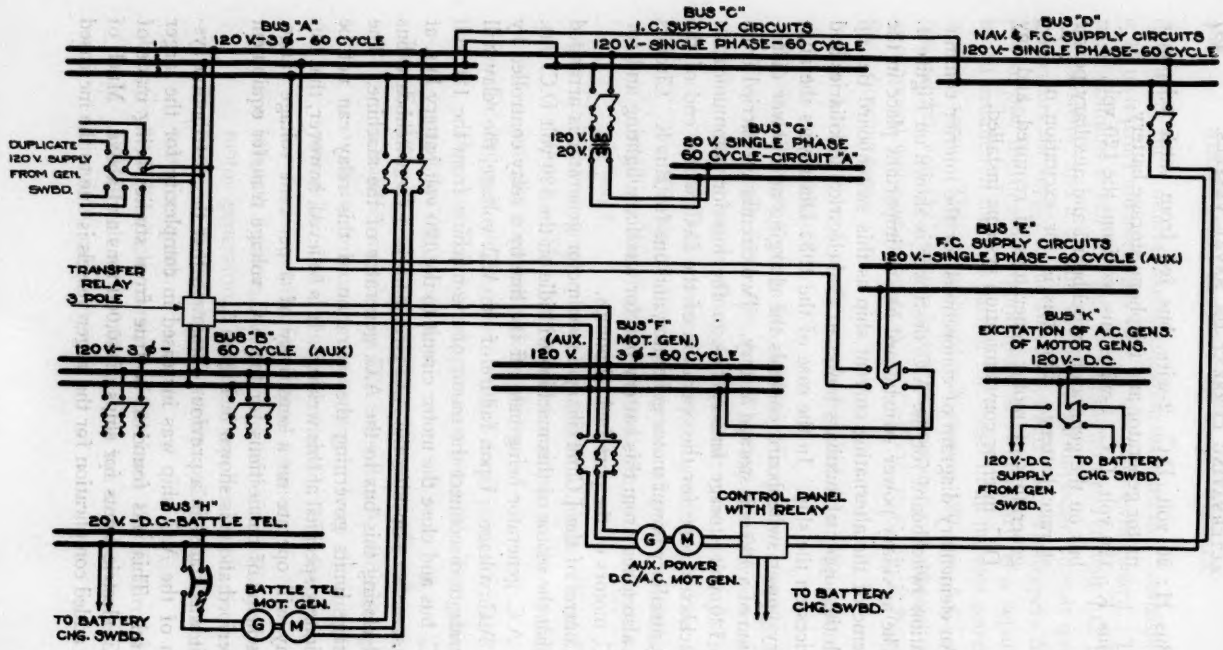


FIG. 4.—ELEMENTARY DIAGRAM CONNECTIONS FOR I. C. SWITCHBOARD FOR A. C. SHIP.

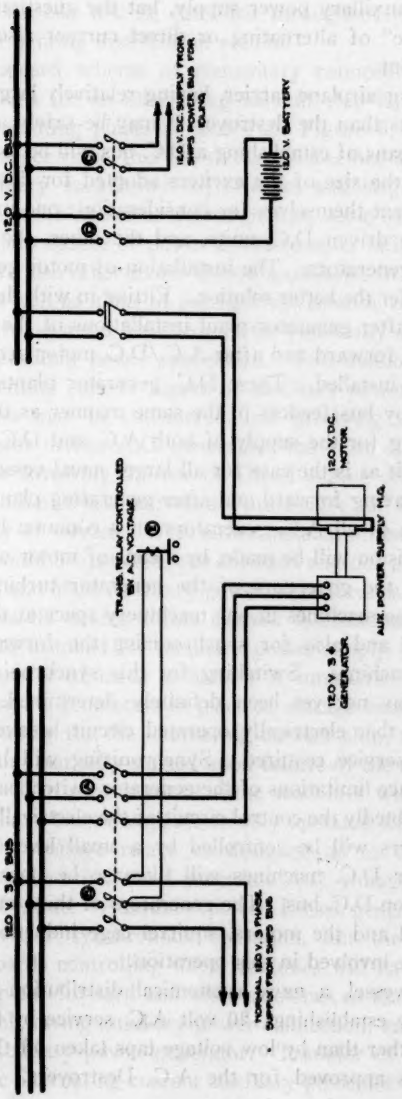

120 V. D.C. BUS
120 V. D.C. SUPPLY FROM SHIP'S POWER BUS FOR EXCITING
120 V. BATTERY
120 V. D.C. MOTOR
TRANS. RELAY CONTROLLED BY A.C. VOLTAGE
120 V. 3ϕ GENERATOR
AUX. POWER SUPPLY M.G.
120 V. 3ϕ BUS
NORMAL 120 V. 3 PHASE SUPPLY FROM SHIP'S BUS
④ CONTACTORS OPEN AND ⑤ CONTACTORS CLOSED WHEN 120 V. 3ϕ SHIP'S BUS VOLTAGE IS NORMAL.
③ CONTACTORS CLOSED AND ⑤ CONTACTORS OPEN WHEN 120 V. 3ϕ SHIP'S BUS VOLTAGE IS NORMAL.
UPON FAILURE OF 120 V. 3ϕ SHIP'S BUS VOLTAGE ③ CONTACTORS AND ⑤ CONTACTORS OPEN AND ④ CONTACTORS AND ⑤ CONTACTORS CLOSE.
OPERATION OF CONTACTORS CONTROLLED BY VOLTAGE RELAY ②

FIG. 5.—CONNECTIONS FOR BUS TRANSFER FOR AUXILIARY POWER MOTOR GENERATOR.

requirements for auxiliary power supply, but the question of the "proportionate use" of alternating or direct current also enters into the development.

For the cruiser or airplane carrier, having relatively larger D.C. power requirements than the destroyer, it may be safely assumed that some other means of establishing a D.C. bus will be used than that of increasing the size of the exciters adopted for destroyers. Two methods present themselves for consideration; one involving the use of turbine driven D.C. units, and the other, the use of A.C./D.C. motor generators. The installation of motor generator units appears to offer the better solution. Fitting in with the duplicate forward and after generator plant installations of the cruiser or airplane carrier, forward and after A.C./D.C. motor generators would likewise be installed. These D.C. generator plants would be interconnected by bus feeders in the same manner as the A.C. machines, providing for the supply of both A.C. and D.C. power from forward or aft as is the case for all larger naval vessels.

For the vessel having forward and after generating plants, synchronous operation of all A.C. generators on a common bus will be required. Provision will be made, by means of motor operated remote control on the governors of the generator turbines, for synchronizing of the machines in one machinery space at the generator switchboard and also for synchronizing the forward and after groups of machines. Switching for this synchronizing of A.C. generators has not yet been definitely determined. It is believed, however, that electrically operated circuit breakers will best perform the service required. Synchronizing will be done automatically if space limitations of the generator switchboard permit. If not, undoubtedly the control circuit of the electrically operated circuit breakers will be controlled by a small lever switch. Forward and after D.C. machines will likewise be operated in parallel on a common D.C. bus. The generators of these machines being shunt wound and the motors, squirrel cage induction type, no difficulty will be involved in this operation.

For the larger vessel, a more economical distribution system will be obtained by establishing 120 volt A.C. service by the use of transformers rather than by low voltage taps taken off the generator armature as approved for the A.C. Destroyers. Trans-

former stations will be provided throughout the vessel, as best suits the lighting distribution system.

A proposed scheme of elementary connections for generator switchboard for an alternating current ship having forward and after generating plants is shown on Figure 6. It will be noted for the larger vessel that the voltage is specified as 450 volts.

Machinery installations of cruisers and larger vessels require the installation of motor driven pumps, some of which must have an auxiliary source of power available upon failure of the voltage. There are also other motor driven auxiliaries in other parts of the vessel as well as items of the lighting, interior communication and fire control systems, that require the continuity of service afforded by the auxiliary power connection. Two methods of establishing an alternating current source of auxiliary power supply for these auxiliaries may be considered, one involving the installation of D.C. motor driven A.C. generators fed from storage battery and the other involving the use of Diesel engine driven A.C. generators. Both of these methods are practicable and both offer an equal reliability of service. With either installation two auxiliary power plants would be installed, one forward and one aft controlled from forward and after auxiliary power switchboards.

On vessels having direct current installations, storage battery plants are installed for auxiliary power supply. In these installations transfer of loads to battery is automatic. With the use of Diesel engine generators in place of storage batteries for auxiliary power supply for the alternating current ship, the Diesel engines will be automatically started upon failure of the ship's service A.C. voltage. The generators of these machines will be connected to the auxiliary bus through electrically operated circuit breakers controlled by voltage relays so designed as to close these circuit breakers when the Diesel generator A.C. voltage rises to a predetermined value. Auxiliary power Diesel generators will not be arranged for synchronous operation, but the auxiliary power switchboards controlling these machines will have duplicate buses fed from the forward and after machines so that any auxiliary on the vessel, having auxiliary power connection, may be connected to either auxiliary power generator. Transfer of auxiliaries to and from the alternating current auxiliary power bus will be automatic

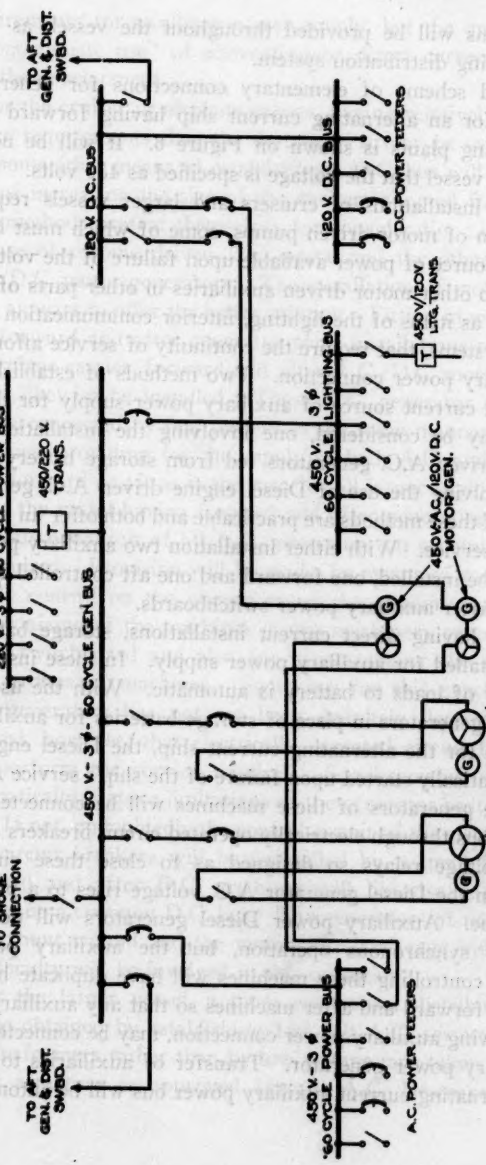

TO AFT GEN. & DIST. SWBD.
120 V. D.C. BUS
120 V. D.C. BUS
DC. POWER FEEDERS
450V/120V LTG. TRANS.
450 V. 3 ϕ 60 CYCLE LIGHTING BUS
450 V.AC./120V.D.C. MOTOR GEN.
450 V-3 ϕ 60 CYCLE POWER BUS
450/220 V. TRANS.
220 V-3 ϕ - 60 CYCLE POWER BUS
450 V - 3 ϕ - 60 CYCLE GEN. BUS
450 V-3 ϕ ALTERNATOR 120 V-D.C. EXCITER
450 V-3 ϕ ALTERNATOR 120 V-D.C. EXCITER
TO SHORE CONNECTION
450 V - 3 ϕ 60 CYCLE POWER BUS
A.C. POWER FEEDERS
TO AFT GEN. & DIST. SWBD.

FIG. 6.—ELEMENTARY DIAGRAM CONNECTIONS FOR GENERATOR SWITCHBOARD FOR A. C. SHIP HAVING FORWARD AND AFTER GENERATOR PLANTS.

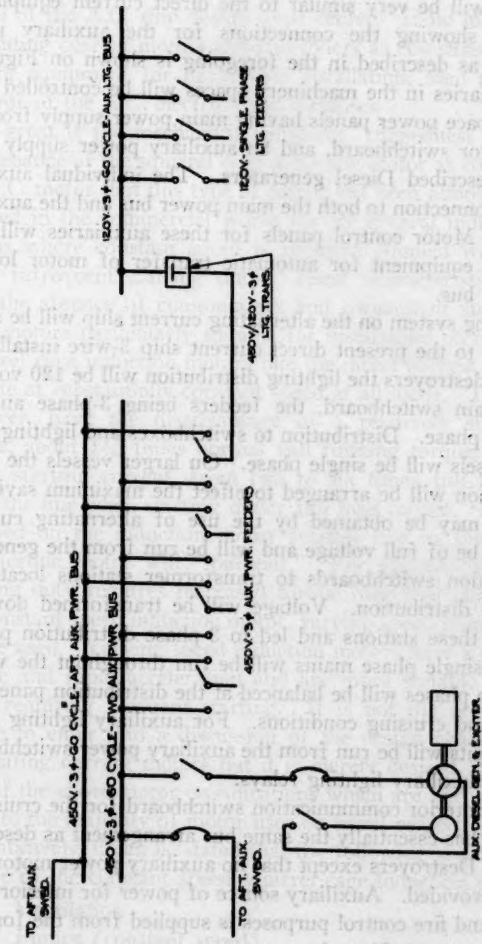

TO AFT. AUX. SWBD.
450V-3ϕ-60 CYCLE-AFT. AUX. PWR. BUS
450V-3ϕ-60 CYCLE-FWD. AUX. PWR. BUS
450V-3ϕ AUX. PWR. FEEDERS
TO AFT. AUX. SWBD.
120V-3ϕ-60 CYCLE-AUX. LTG. BUS
120V-SINGLE PHASE LTG. FEEDERS
450V/120V-3ϕ LTG. TRANS.
AUX. DIESEL GEN. & EXCITER 450V-3ϕ-60 CYC.

FIG. 7.—ELEMENTARY DIAGRAM CONNECTIONS FOR AUXILIARY POWER SWITCHBOARD.

by individual bus transfer panels similar to the arrangement on D.C. ships. The characteristics of the alternating current transfer equipments will be very similar to the direct current equipments. A diagram showing the connections for the auxiliary power switchboard as described in the foregoing is shown on Figure 7.

The auxiliaries in the machinery spaces will be controlled from machinery space power panels having main power supply from the local generator switchboard, and an auxiliary power supply from the above described Diesel generators. The individual auxiliary will have a connection to both the main power bus and the auxiliary power bus. Motor control panels for these auxiliaries will have bus transfer equipment for automatic transfer of motor load to the auxiliary bus.

The lighting system on the alternating current ship will be somewhat similar to the present direct current ship 3-wire installation. On the new destroyers the lighting distribution will be 120 volt fed from the main switchboard, the feeders being 3-phase and the mains single phase. Distribution to switchboxes and lighting units on these vessels will be single phase. On larger vessels the lighting distribution will be arranged to effect the maximum saving in weight that may be obtained by the use of alternating current. Feeders will be of full voltage and will be run from the generator and distribution switchboards to transformer stations located to best suit the distribution. Voltage will be transformed down to 120 volts at these stations and led to 3-phase distribution panels, from which single phase mains will be run throughout the vessel. Loads on the phases will be balanced at the distribution panels for both battle and cruising conditions. For auxiliary lighting units, 120 volt circuits will be run from the auxiliary power switchboards to the usual auxiliary lighting relays.

The main interior communication switchboard for the cruiser or larger vessel has essentially the same bus arrangement as described for the 1932 Destroyers except that no auxiliary power motor generators are provided. Auxiliary source of power for interior communication and fire control purposes is supplied from the forward auxiliary power switchboard.

Motors and their control are the principal items of electric plant equipment for the alternating current ship for which simplification of design, decrease in weight and greater efficiency in

operation and maintenance may be expected. In fact it is around the question of motors and their control that discussion of the merits of the A.C. ship design must center. If no merit exists in alternating current motors and controllers as compared with direct current equipment for naval ship installation, then all argument in favor of the A.C. electric plant collapses.

So great is the mechanical simplicity, ruggedness and ease of maintenance of the squirrel cage induction motor as compared with the D.C. motor that this type of motor has been almost universally adopted in the commercial world for all general industrial applications where constant speed is required coupled with comparatively infrequent starting duty. These characteristics combined with the absence of commutators and consequent sparking make these motors particularly suitable for installation on naval vessels. Squirrel cage induction motors like the direct current shunt motor operate at substantially constant speed at all loads. These motors, however, have the disadvantage of a relatively small starting torque, about 1¾ times normal full running torque, and draw a heavy starting current, from two to six times full load current, from the line when starting under full load torque. If the torque imposed on the induction motor reaches two to four times full load torque motor will "pull-out." For applications where the starting duty requires increased torque over that obtainable with the constant speed induction motor, varying speed induction motors can be used. Varying speed induction motors have a high starting torque and rapid acceleration and are more suitable for applications requiring frequent starting. It is not the purpose of this article to enter into a discussion of the design characteristics of alternating current motors but it is merely desired to make mention of the above motor classifications which are particularly applicable to the majority of motor uses on naval vessels.

Tentatively, it can be stated that the following shipboard motors will be of the constant speed squirrel cage induction type:

Ventilation
Pumps (constant speed)
Shop machinery
Laundry machinery
Galley and pantry equipment
Motor-generators

Likewise, the following motors may be of the varying speed induction type:

Air compressors.
Hydraulic plant
Crane
Laundry extractor
Shaft turning

For some motor applications, such as for turrets, windlass, boat crane hoists, winches, and elevators, special study will have to be made to determine the most suitable type of motor to be used. Special types of motors or possibly motors with speed reducers may be used for slow speed auxiliaries. For certain variable speed applications slip-ring or wound-rotor induction motors having external starting resistances may be used. These motors operate with characteristics similar to direct current motors having resistance in the armature circuit. In other variable speed applications hydraulic speed control may be used in combination with constant speed induction motors.

Due to the inherent low power factor of under loaded induction motors, it is extremely important that the horsepowers of these motors as applied to the alternating current ship be not in excess of the horsepower requirements of the driven auxiliary. In the case of the direct current ship the installation of excess motor horsepower is of no particular importance other than the slight increase in weights and some decrease in the operating efficiency of the plant. However, for the alternating current ship it is necessary that the size of motors be restricted to the actual requirements of the driven auxiliary in order not to "pull-down" the power factor of the electric plant. The use of synchronous motors for the correction of power factor is being considered. Motor applications on shipboard to which the synchronous motor can be applied for the purpose, however, are limited.

Taking up the subject of motor control equipment, it is contemplated that by the use of alternating current motors there will be a marked decrease in the amount of motor control apparatus that the vessel must carry. Navy Department specification 17-M-9 requires that all direct current motors, other than small motors of 2 horsepower or less which may be safely thrown directly in

the line without previous acceleration, be provided with controllers for starting duty. From actual direct current ship installations it is found that the majority of direct current motors down to ratings of 1 horsepower are provided with starting panels.

Squirrel cage induction motors may be safely started by throwing them directly on the line. This method is ordinarily used only for small motors less than 10 horsepower because of the excessive starting currents of these motors creating too heavy a disturbance on the line. For the alternating current naval vessel it is probable that across the line starting will be limited to motors of 5 horsepower and less in order not to affect the voltage regulation of the ship's generators. For motors above 5 horsepower it is proposed to limit the in-rush current on the generators by the use of compensating control equipment.

An analysis of the controller equipment of the direct current motor installation of one of the new 10,000 ton cruisers shows that, of approximately 210 motors of all horsepowers installed, about 10 per cent have across the line switches for starting, the remaining motors having control panels with starting resistance. With an alternating current installation the number of motors up to and including 5 horsepower which can be started with across-line starters would be approximately 65 per cent of the total installed leaving about 35 per cent of motors on the vessel requiring compensating control equipment. Thus the saving in bulkhead space, wiring, weight, spare parts and maintenance resulting from the use of alternating current control equipment is considerable.

The wiring installation for the alternating current ship will not differ greatly from the direct current ship. It is proposed to use 3-phase, 3 conductor cable for 3-phase circuits up to and including a conductor size of 400,000 circular mils. Above 400,000 circular mils single conductor cables will be used. Where single or multiple cables are installed for the individual phases it is proposed that these cables be installed with a twist having a lay of 20 cable diameters, approximately, to neutralize inductive effect. Single conductor alternating current cables will not be grouped in the same hangers with direct current cables. It is also desirable to separate alternating current cable runs where a difference in frequency exists.

It is known that electrolysis, due to stray currents from the direct current installation, exists on naval vessels, sometimes with serious results. Electrolytic action from alternating current is known to be generally less than 1 per cent of that which would be produced by a corresponding direct current. The adoption of the alternating current electric plant will remove this problem from the consideration of the operating personnel.

Present standard designs of electrical fixtures, boxes, switches and other various fittings will not change appreciably for the 230 volt alternating current destroyers. New designs of distribution panels, and feeder distribution boxes, will, however, be necessitated by the adoption of the 3-phase wiring system, as these designs have been in the past 2-pole direct current. For the 450 volt ship, a more extensive redesign of electrical fittings will be required, due both to the use of the 3-phase system and to the increase in voltage. The increase in length of 450 volt fuses over 250 volt fuses will require designs of fuse boxes to be changed to accommodate the larger fuses. This redesign, will however, not be serious as only the power system circuits will be involved.

The question of comparative weights of the alternating current ship and the direct current ship electric plant installation is one of the most vital concerned with the proposed alternating current ship development. The first study made of the actual weights of the electric plant (excluding motors and controllers) of one of the 10,000 ton cruisers in comparison with the estimated weights of a tentative alternating current plant showed the weight of the alternating current installation to be close to 20 per cent greater than the direct current installation. This study was made on the basis of the alternating current installation being identical with the direct current installation except for inherent changes between the two installations. A further study of a new class of vessel for both alternating current and direct current installations taking advantage of any weight saving possibilities in the alternating current installation, such as increased voltage and the omission of motor generators, etc., showed the weight of the alternating current installation approaching more nearly the direct current installation. These weight figures cover the generating plant, the power and

lighting wiring systems, the auxiliary power installation, the interior communication and fire control systems, etc., but do not include motors and control equipment, which items are not included as "electrical weights." It is realized that a true picture of the alternating current ship weights should include all motors and current consuming devices and their control equipment proposed for the vessel, particularly as it is in these items that a reduction in weight must be found to compensate for increases in weights of other electric plant equipment. On the basis of present catalogued weights of commercial alternating current motors it appears that the weights of these motors will run from 30 per cent less for small motors to 50 or 60 per cent less for large motors. It is difficult to form any definite comparison for alternating current and direct current controller weights owing to the considerable difference existing in the application of these controllers referred to in a previous paragraph. It is accepted, of course, that alternating current controller weights will be much less than direct current controller weights. If arbitrary values of weight saving of 125 pounds per motor and 75 pounds per controller were assumed, on the basis of the present electric plant installation of a 10,000 ton cruiser a saving in weight due to the installation of alternating current motors and controllers of over 10 tons may be reasonably expected. Such a figure may be sufficient to offset any increased weight arising from any other part of the alternating current installation.

It is confidently expected, and the expectation seems to be justified by development to date on the 1932 Destroyers, that with the field of alternating current motor and generator design opened up to the possibilities of weight saving, that a considerable saving in this respect will result from improved design of such equipment. It is also expected that additional means of saving weight will be found in the wiring installation and other items of the electric plant installation as the development of the alternating current ship proceeds. With these developments and with the expected reduction in weight due to reduced quantities of spare parts, it appears certain that the weight of alternating current electrical installation will compare favorably with that of the direct current ship. If this is true then an appreciable improvement in the field of electrical design as applied to naval ships will have been made.

UNITED STATES LINES LINER S. S. *MANHATTAN.*

DESCRIPTION AND TRIALS.

By M. W. Torbet, B. Mar. Eng., Member.

The *Manhattan,* delivered on July 27, 1932, is the first of two sister ships to be placed by the United States Lines in their New York-Hamburg service. The second ship, the *Washington,* was launched at the plant of the New York Shipbuilding Company on August 20, 1932, and will be delivered early in the summer of 1933. See photograph Figure 1.

The *Manhattan* is notable both in that she is the largest merchant vessel so far built in the United States and is the first passenger and cargo vessel to be built in the United States in thirty years expressly for the North Atlantic passenger and cargo trade.

The *Manhattan* and *Washington* are built under the provisions of the Jones-White Act, in accordance with the latest rules of the American Bureau of Shipping, from contract plans approved by the United States Shipping Board and the Navy Department. Both vessels conform to American Bureau of Shipping Classification ✠ A-1 (E). Design plans, calculations, and contract plans were prepared by the New York Shipbuilding Company. Specifications were prepared by the owners. The contract was signed on May 24, 1930.

The dimensions and characteristics of these vessels were decided upon after an extensive study of the trend in the North Atlantic trade. Restricted immigration has greatly reduced the number of third class passengers. The number of passengers going to Europe for touring and educational purposes, on the other hand, is increasing, and creates a demand for a special tourist class accommodation. The recent "cabin" ships have been supplying the demand formerly supplied by second class accommodations on the large fast liners. First class has become associated with high speed. During the winter season cruises to tropical waters have become popular.

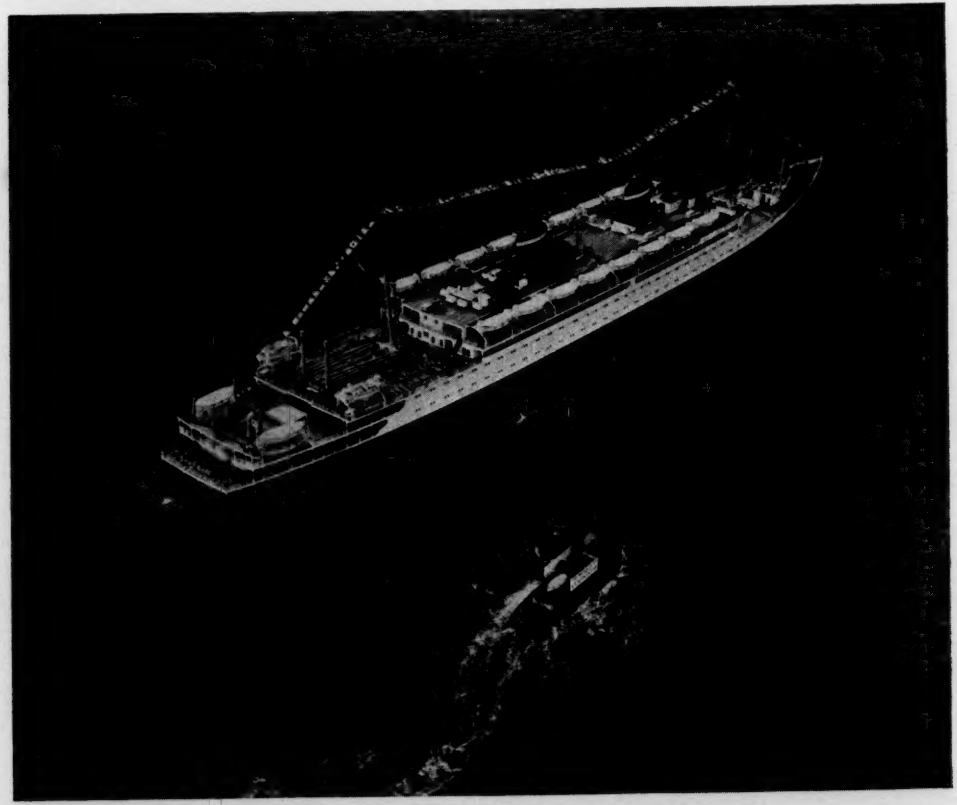


FIG. 1.—S. S. "MANHATTAN" ENTERING NEW YORK HARBOR.

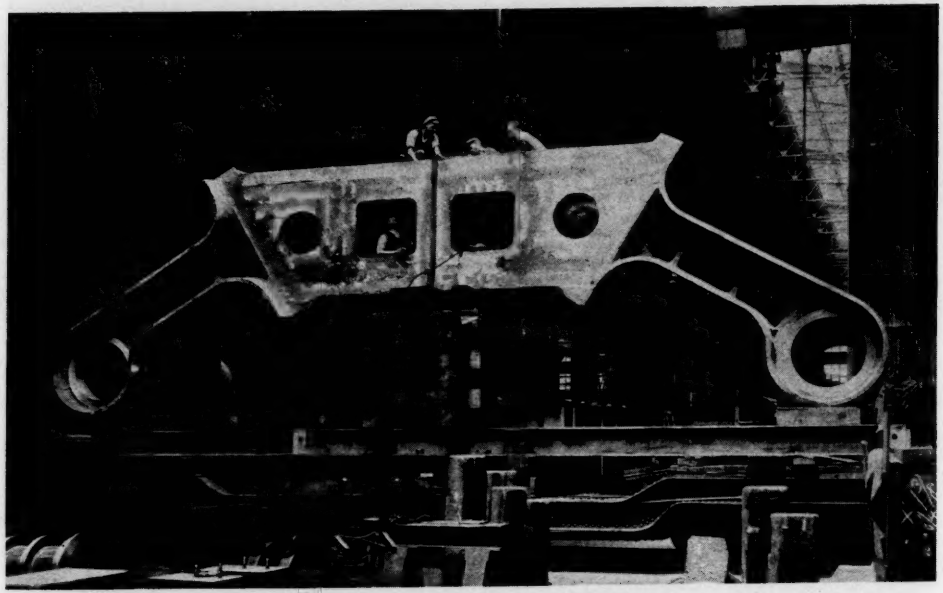


FIG. 2.—THE 73-TON, SPECTACLE FRAME, WITH THE TWO SECTIONS BOLTED TOGETHER IN THE SHOP.

The principal requirements for these vessels provided for primarily cabin class ships with maximum comfort for cabin, tourist, and third class passengers; passenger accommodations arranged to be thrown together into one class for winter cruises; an average service speed of 20 knots; 12,000 tons deadweight; bunkers for round trip voyage New York to Hamburg and return; maximum fuel economy consistent with conservative and proved boiler, turbine, and gear design. As finally developed the principal dimensions and characteristics became as follows:

PRINCIPAL DIMENSIONS AND CHARACTERISTICS.

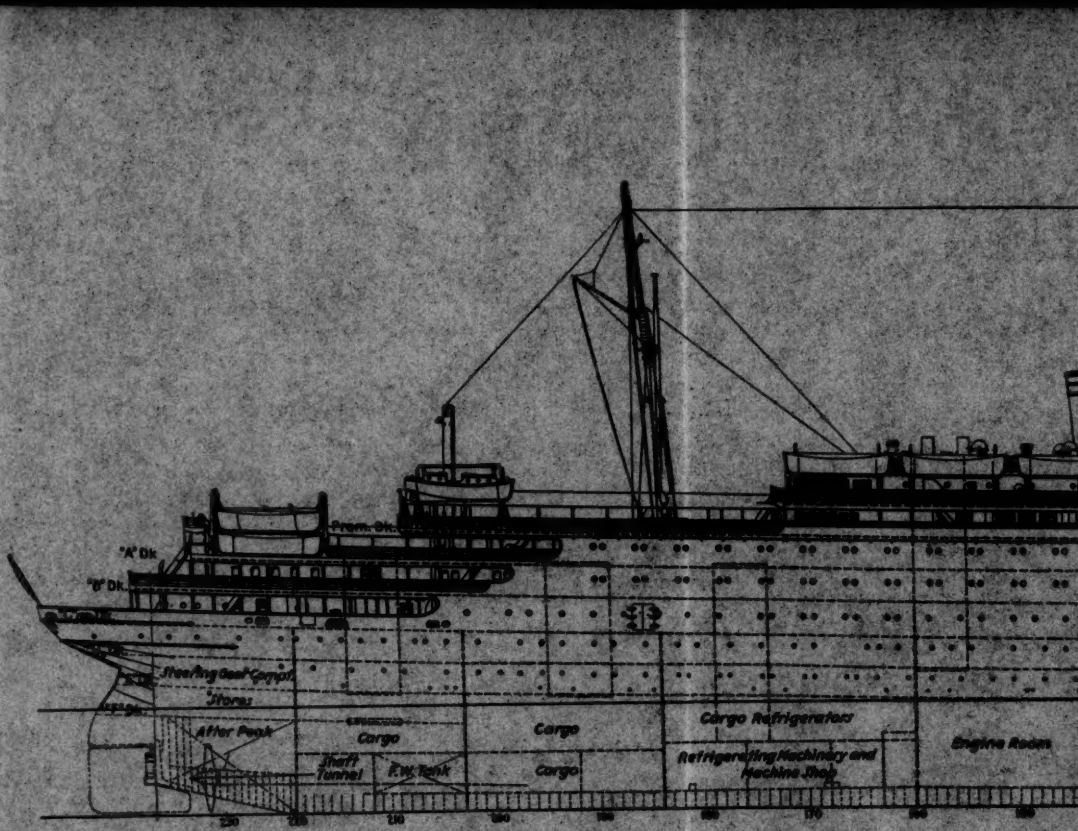
Length, overall, feet	705
Length, on water line, feet	685
Length, between perpendiculars, A.B.S. rules, feet	666
Beam, molded, feet	86
Depth to "C" deck, molded, feet	47
Depth to Promenade deck, feet	75
Camber of "C" deck and above, in 86 feet, inches	6
Designed load draft, feet	30
Designed deadweight on 30 foot draft, tons	12,000
Displacement at legal draft of 30 feet, 8¾ inches, tons	33,500
Gross tonnage, tons	24,289
Net tonnage, tons	13,924
Normal service speed, knots	20
Normal service power, S.H.P.	30,000
General cargo, cubic feet	380,000
Refrigerated cargo, cubic feet	47,000
Mail rooms, cubic feet	19,200
Baggage rooms, cabin, cubic feet	8,380
Baggage rooms, tourist, cubic feet	5,450
Baggage rooms, third class, cubic feet	2,040
Ship's cold stores, cubic feet	16,000
Steward's stores, cubic feet	7,900
Bunkers, at 97 per cent full tanks, 38 cubic feet per ton, tons	4,775
Fresh water—potable, tons	616
Fresh water—ship's use, tons	2,283
Fresh water—reserve feed, tons	680
Fresh water—distilled, for boiler feed, tons	248
Cabin passengers	582
Tourist passengers	461
Third-class passengers	196
Total passengers	1,239
Crew	478

Total passengers and crew	1,717
Contract signed	May 24, 1930
Keel laid	Dec. 6, 1930
Launched	Dec. 5, 1931
Delivered	July 27, 1932

DESCRIPTION OF HULL.

The hull is of the complete superstructure type, without expansion joints. Scantlings were determined by considering the equivalent girder as extending continuously up to the Sun Deck. Mild steel was used throughout. There are nine decks, four exclusively for passengers. Frames are transverse, spaced 36 inches for the greater part of the length, reduced to 24 inches in way of peak tanks by decrements of one inch forward and two inches aft. Eleven main transverse bulkheads provide subdivision complying with the International Convention on Safety of Life at Sea (1929). Fuel oil tank and other secondary bulkheads provide further subdivision. The four forward bulkheads extend watertight to "B" deck, the remainder to "C" deck. The subdivision thus insures that the vessel will remain afloat and stable with three compartments flooded either forward or aft, or four compartments amidships. Wing fuel tanks abreast the two boiler rooms and auxiliary engine room safeguard these spaces against all but severe damage, and with the separation of the two boiler rooms by the auxiliary engine room, one collision cannot damage both firerooms as occurred in the case of the *Malolo*.

With fuel carried amidships only a small change of trim is encountered between departure and arrival conditions. Of the nine decks, "C," "D" and "E" extend the full length of the vessel; the Sun Deck extends for a length of 284 feet above the Officers' Quarters; the Boat Deck extends for a length of 300 feet above the Promenade Deck; the Promenade Deck, the strength deck for the midship portion of the hull, extends for a length of 405 feet; "A" deck extends for a length of 520 feet; "B" deck extends from the stem to about 3 feet aft the rudder post, a length of about 670 feet; "F" deck extends forward of the machinery space and aft only in way of No. 6 Hold and above the After Peak Tank. Decks "D," "E" and "F" have no camber; "C" deck and decks above have 6 inches camber in 86 feet. "D" and "E" decks have 'tween


"A" Dk.
"B" Dk.
Steering Gear Compt.
Stores
After Peak
Cargo
Shaft Tunnel
F.W. Tank
Cargo
Cargo Refrigerator
Refrigerating Machinery and Machine Shop
Engine Room

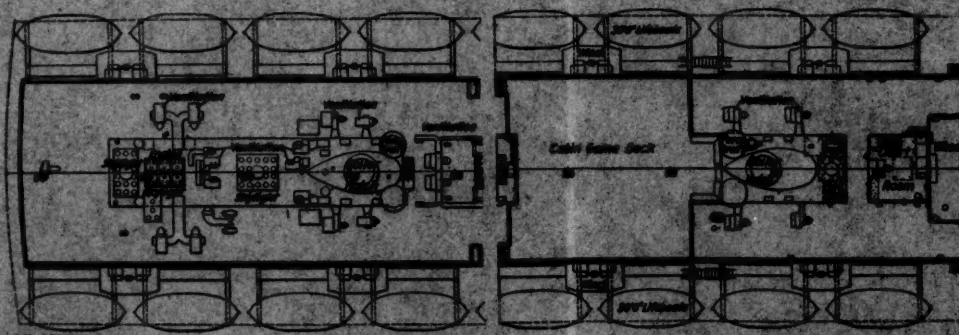

Cabin Space Deck
SUN DECK

Fig. 3.—P

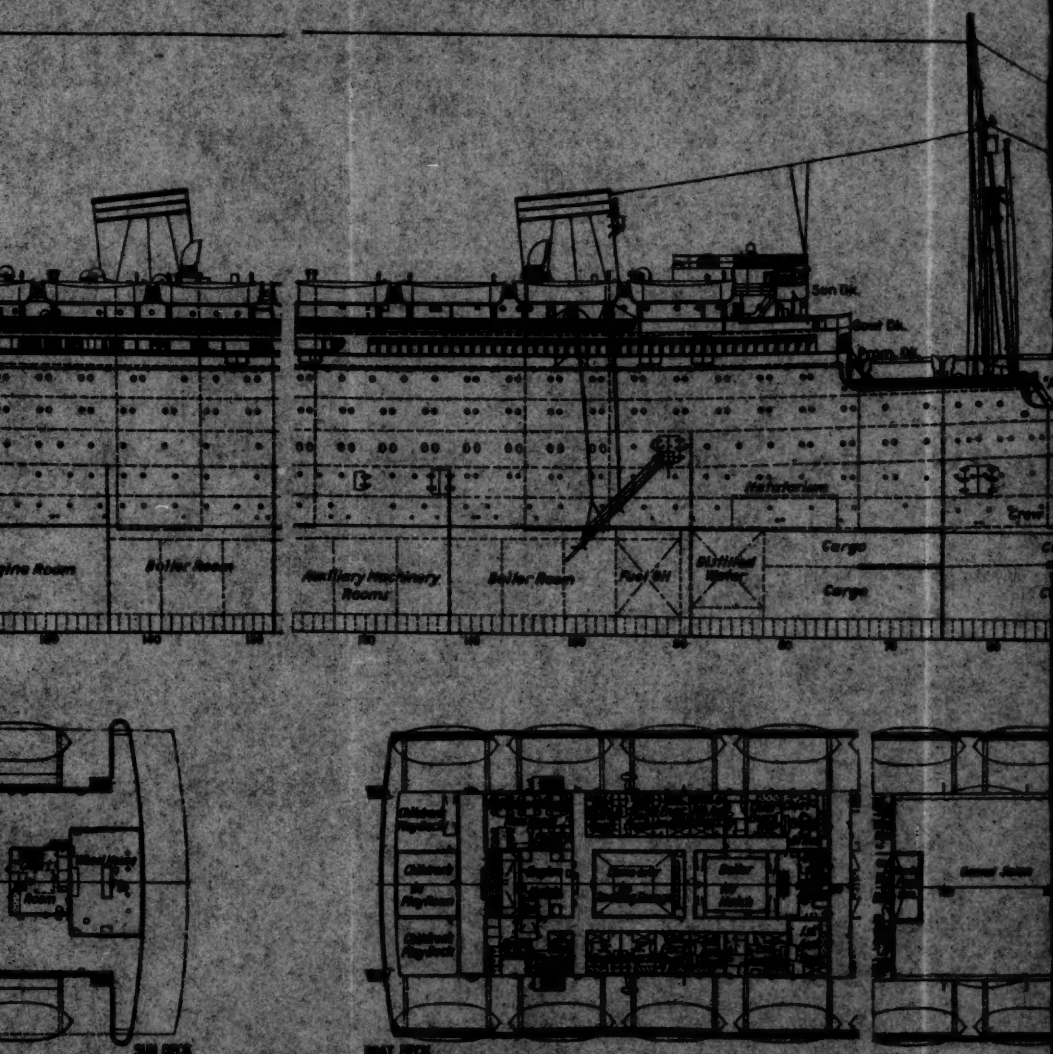


FIG. 1.—PROFILE AND PLANS OF THE SUN AND BOAT DECKS.

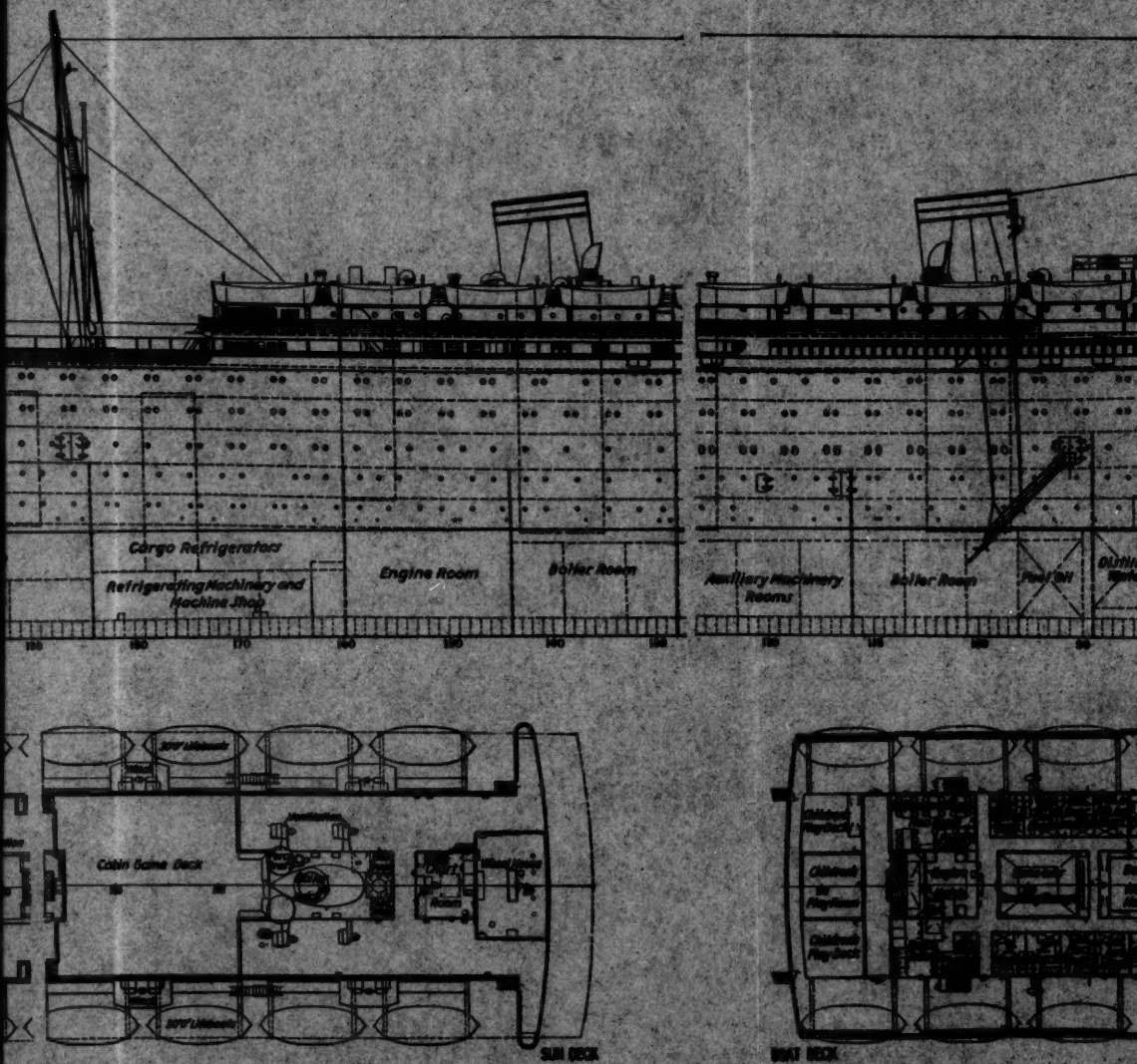
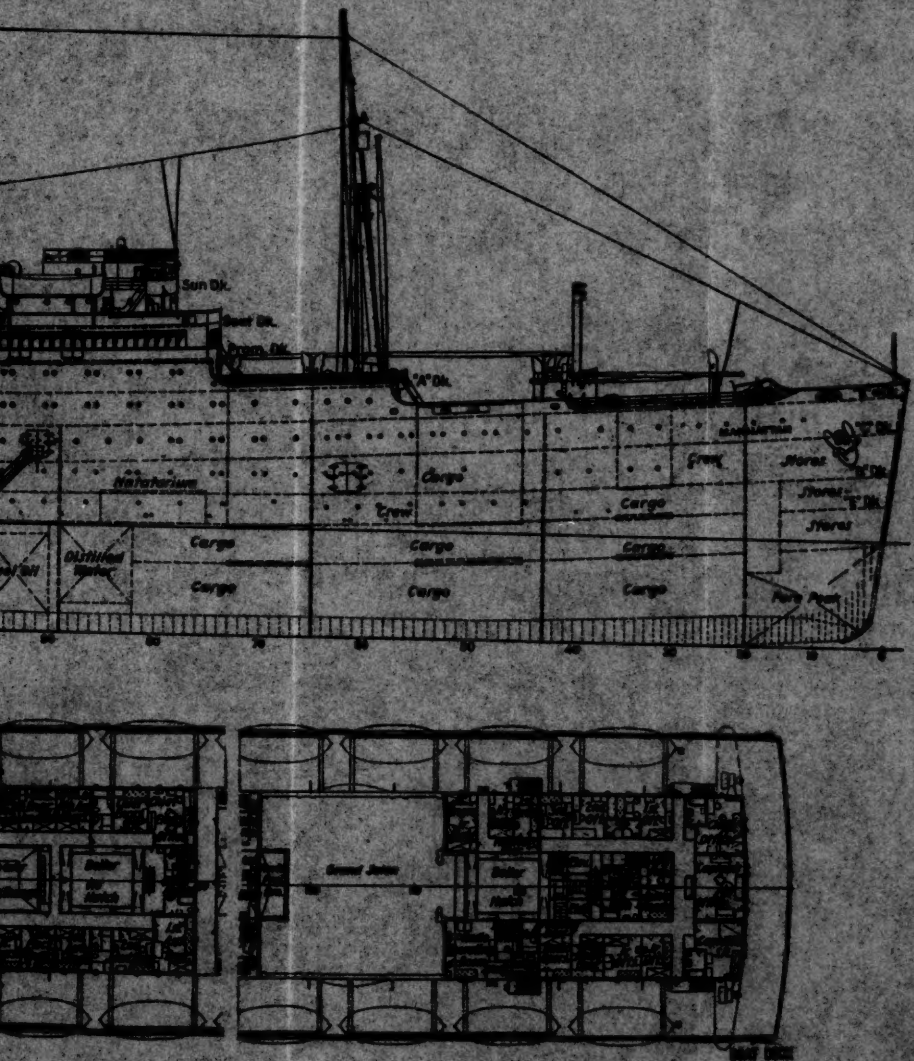


FIG. 3.—PROFILE AND PLANS OF THE SUN AND BOAT DECKS.


Sun Dk.
Boat Dk.
Promen. Dk.
"A" Dk.
Refrigerating
Cargo
Crew
Cargo
Crew
Cargo
Stores
Stores
Stores
Cargo
Cargo
Cargo
Cargo
Cargo
Cargo
Distilled Water
Fore Peak

Engine Room
Boiler Room
Boiler Room
Engine Room
Boiler Room
Boiler Room

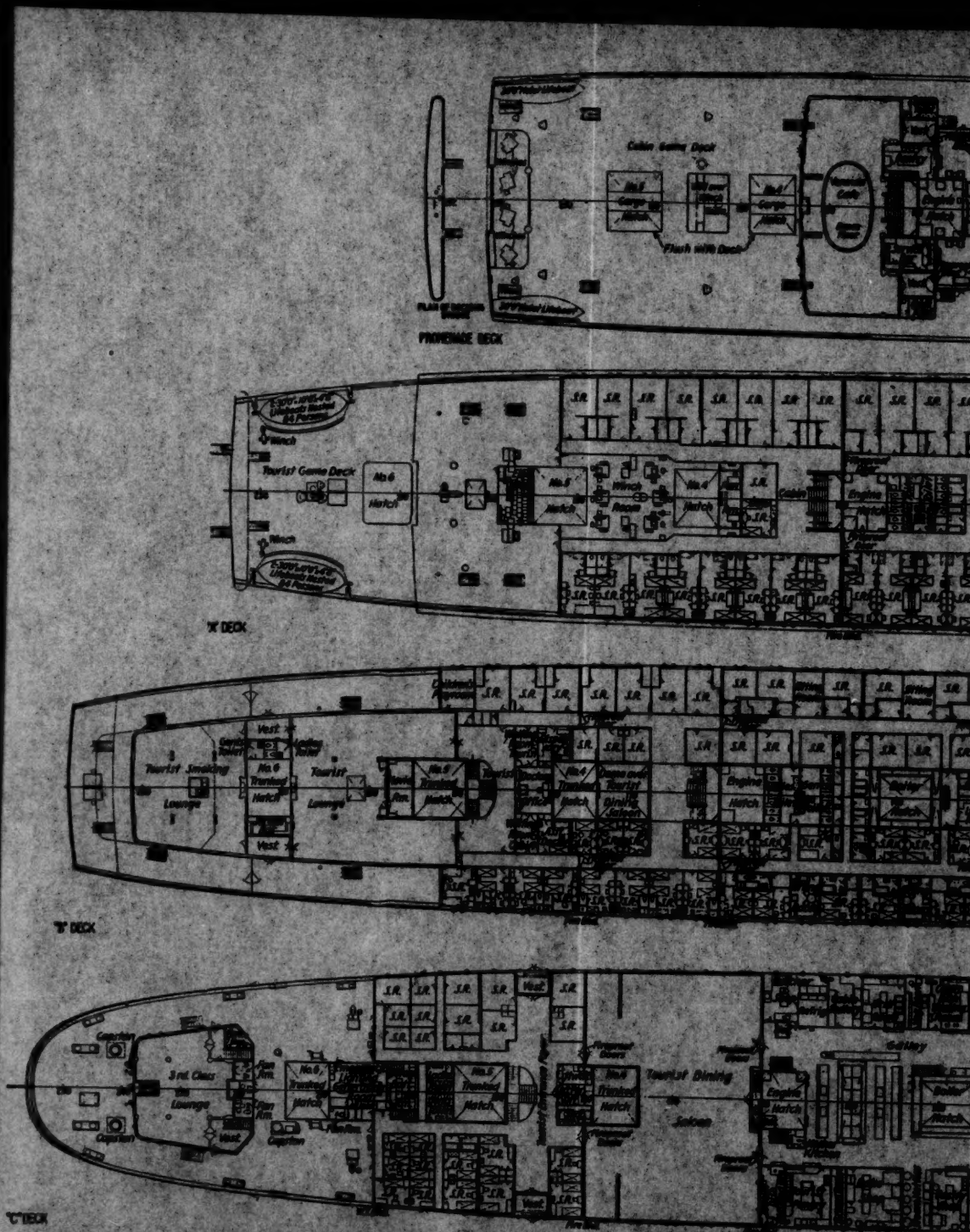


FIG. 4.—PLANS OF THE P

...ED BY THE PROMENADE AND "A," "B," AND "C" DECKS.

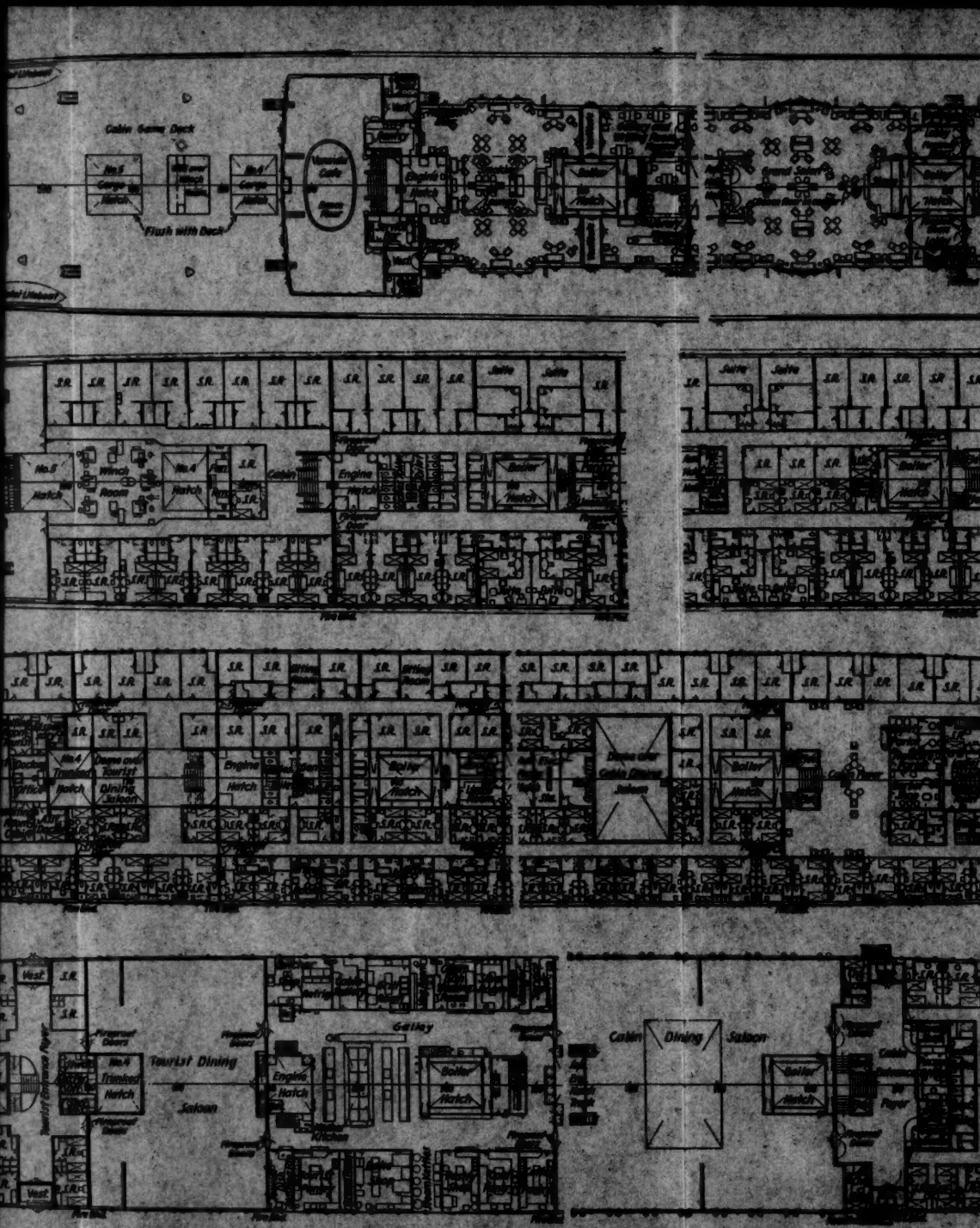


FIG. 4.—PLANS OF THE PROMENADE AND "A," "B," AND "C" DECKS.

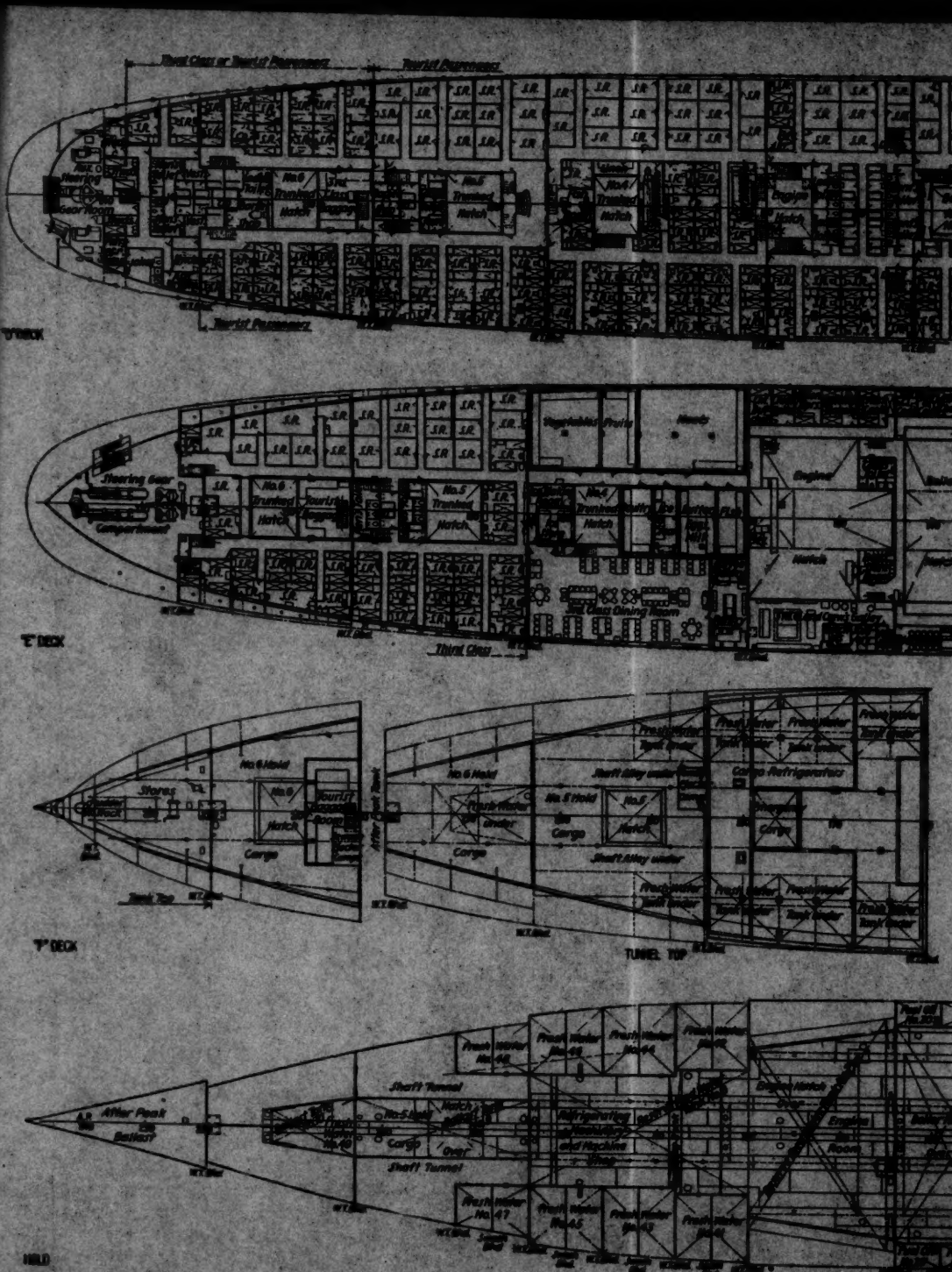
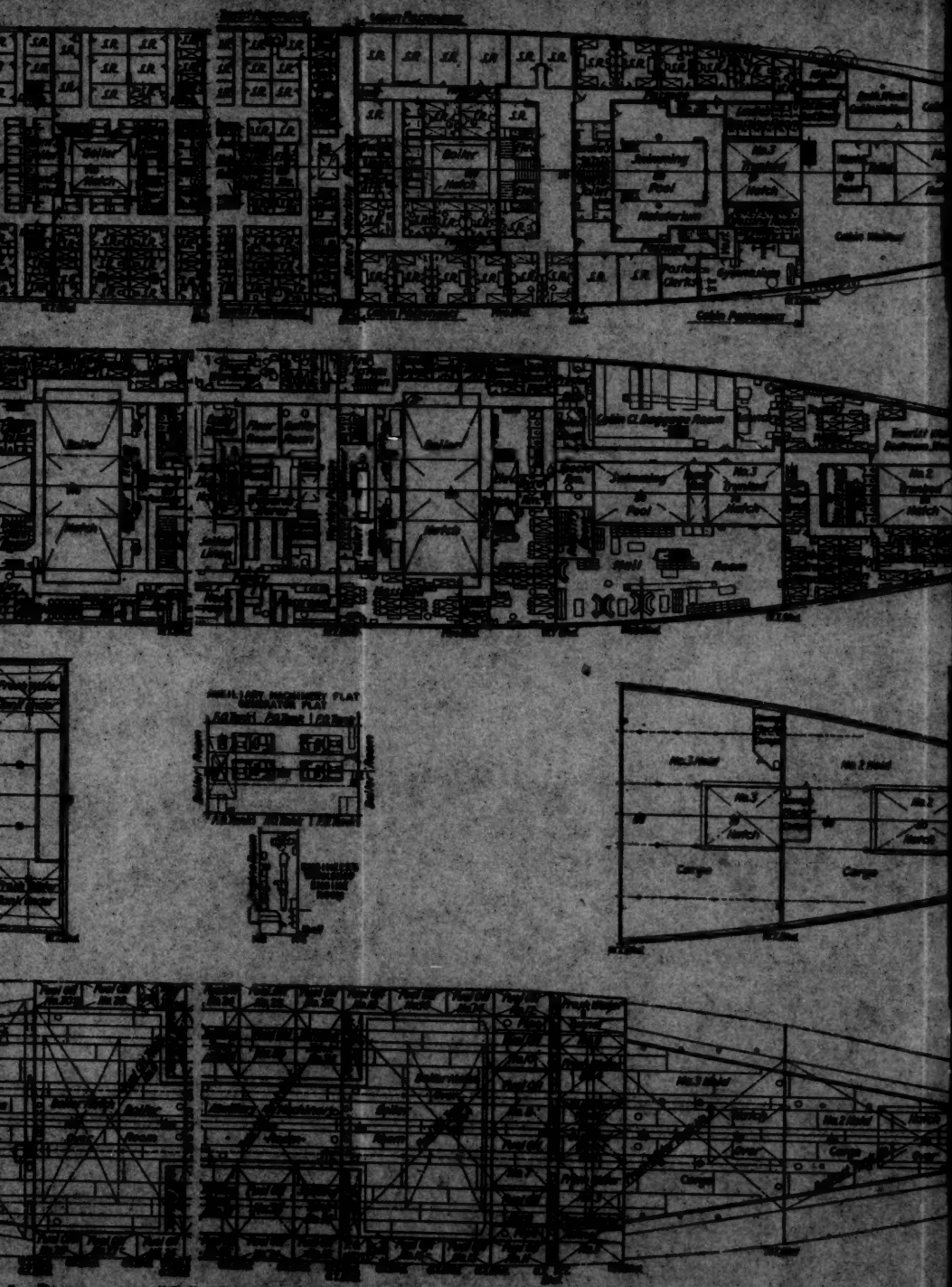

Third Class or Tourist Passengers
Tourist Passengers
Tourist Passengers
Third Class
Steering Gear Compartment
No.6 Trunking Hatch
No.5 Trunking Hatch
No.4 Trunking Hatch
Engine
Boiler
Third Class
Stores
No.6 Hold
No.6 Hatch
Cargo
Tourist Passengers
After Peak Tank
Tank Top
No.5 Hold
No.5 Hatch
Shaft Alley under
Cargo Refrigerator
Tunnel Top
After Peak Ballast
No.5 Hold
Cargo
Shaft Tunnel
Refrigerating and Machine Space
Engine Room
Boiler Room
Shaft Tunnel

FIG. 1.—PLANS



5.—Plans of the "D," "E," and "F" Decks and Hold.

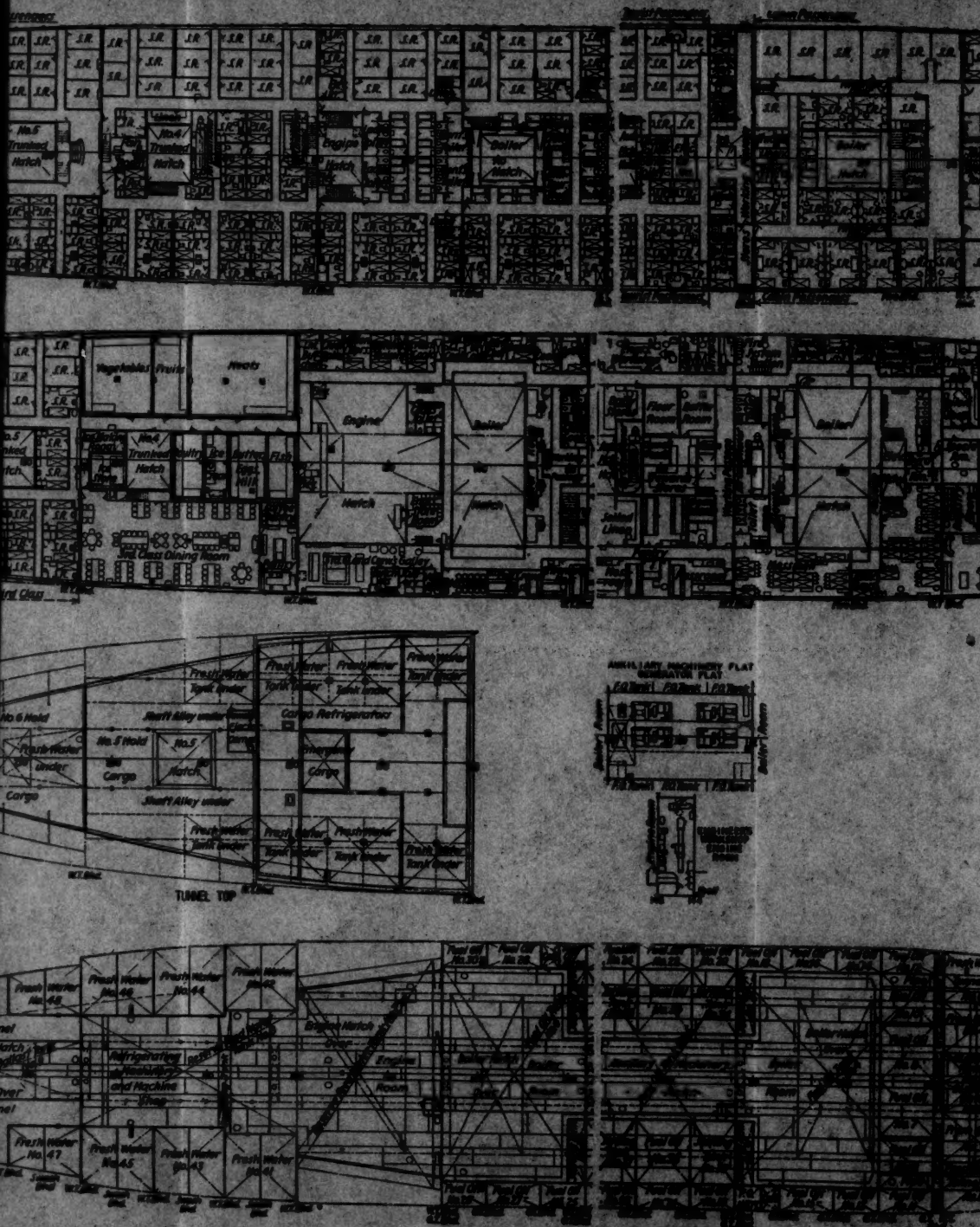
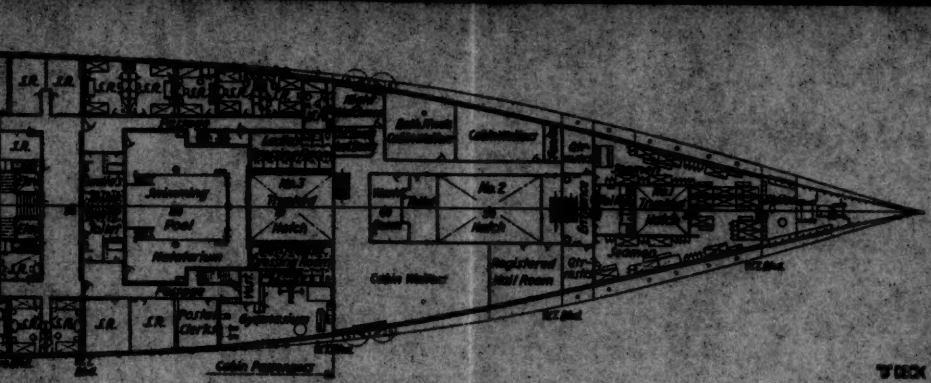
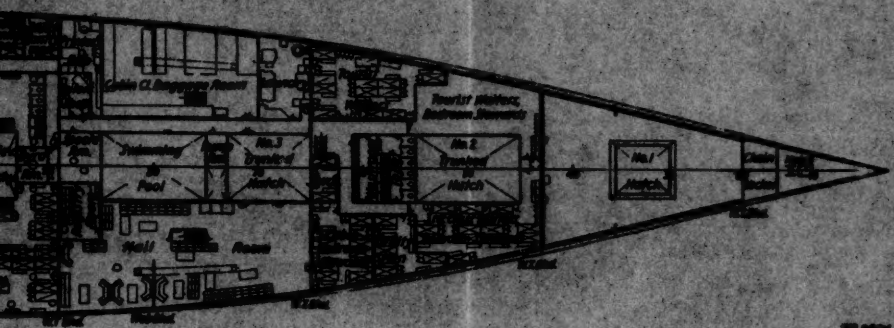


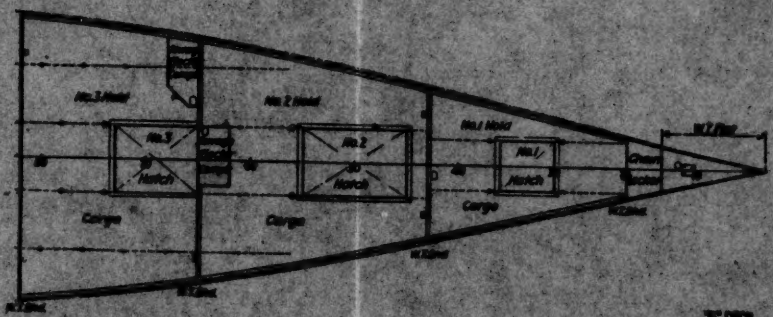
FIG. 5.—PLANS OF THE "D," "E," AND "F" DECKS AND HOLD.



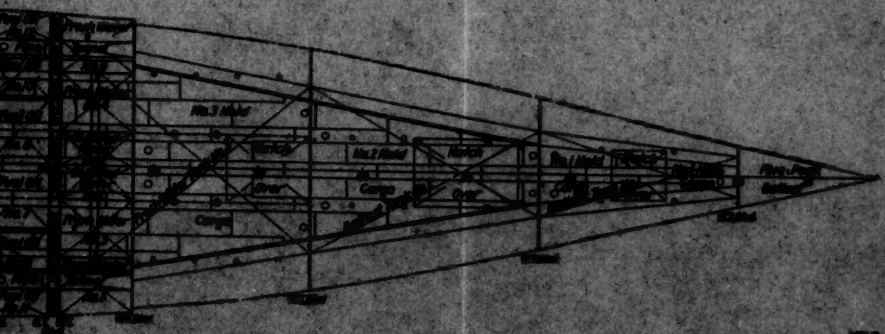
T DECK



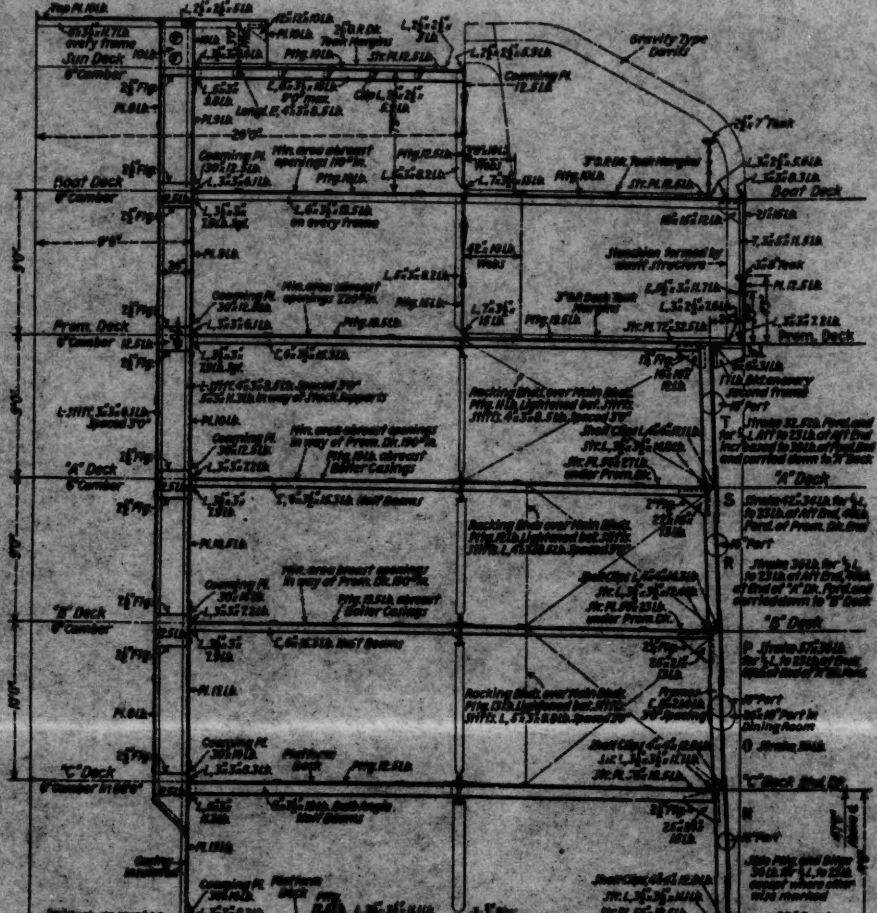
T DECK



T DECK



MAIN


Gravity Type Davits
Boat Deck
Prom. Deck
"A" Deck
"B" Deck
"C" Deck
Sun Deck
Float Deck
Coaming Pl. 12.5 lb.
Coaming Pl. 20 lb.
Coaming Pl. 30 lb.
Shelter Deck
Bulkhead formed by shell structure
Rocking Shelf over Main Shelf
Shelf Clips
Str. Pl. 8/16 lb. under Prom. Str.
Str. Pl. 8/16 lb. under Prom. Dk.
Str. Pl. 7/16 lb.
Str. Pl. 10/16 lb.
Str. Pl. 11/16 lb.
Str. Pl. 12/16 lb.
Str. Pl. 13/16 lb.
Str. Pl. 14/16 lb.
Str. Pl. 15/16 lb.
Str. Pl. 16/16 lb.
Str. Pl. 18/16 lb.
Str. Pl. 20/16 lb.
Str. Pl. 22/16 lb.
Str. Pl. 24/16 lb.
Str. Pl. 26/16 lb.
Str. Pl. 28/16 lb.
Str. Pl. 30/16 lb.
Str. Pl. 32/16 lb.
Str. Pl. 34/16 lb.
Str. Pl. 36/16 lb.
Str. Pl. 38/16 lb.
Str. Pl. 40/16 lb.
Str. Pl. 42/16 lb.
Str. Pl. 44/16 lb.
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Str. Pl. 64/16 lb.
Str. Pl. 66/16 lb.
Str. Pl. 68/16 lb.
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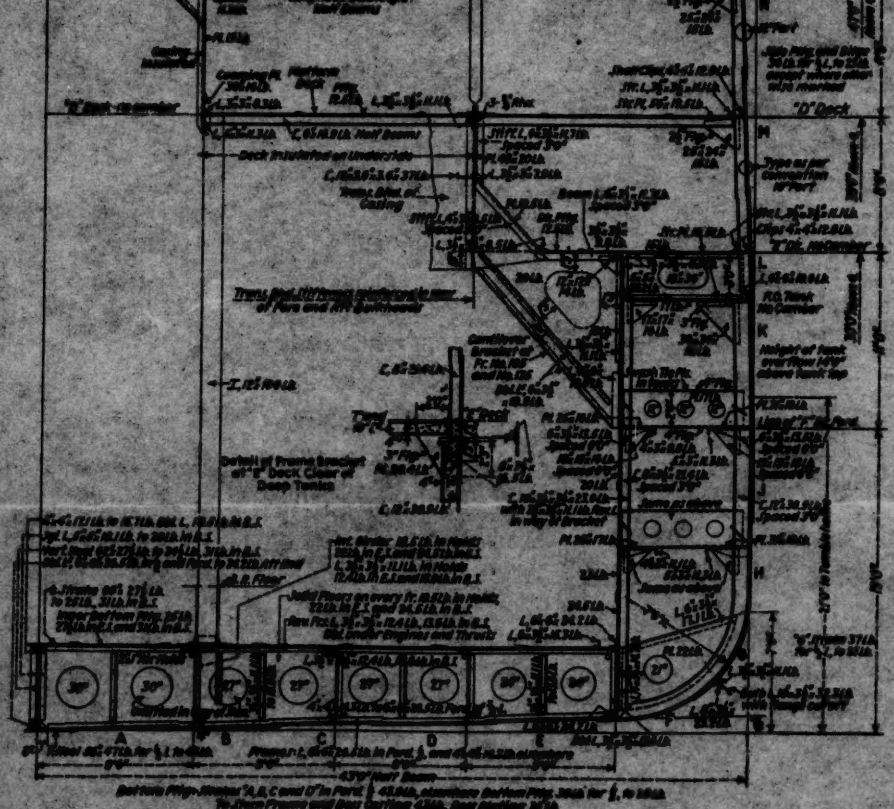


FIG. 8.—MIDSHIP SECTION.

deck heights of 8 feet 6 inches; "C" deck, 10 feet; "B" deck, "A" deck and the Promenade Deck 9 feet; except that in way of the Palm Court on the Promenade Deck the deck height is increased to 10 feet 6 inches.

The depth of the double bottom is 5 feet 2 inches, except in the forward hold where the depth is increased to 8 feet at the collision bulkhead. The stem has a cast steel fore-foot provided with an extension upon which to rig paravane gear. The upper section of the stem is of rolled steel, the forward edges rounded to prevent cutting of lines. The stern post is of cast steel in three pieces, the total weight being 43 tons. The rudder is of the balanced type of stream line design plated over a cast steel frame in three pieces weighing 28 tons. The rudder stock is 24 inches in diameter with a coupling to the rudder frame at the lower end. The spectacle frame weighing 73 tons was cast in two pieces. See Photograph Figure 2.

GENERAL ARRANGEMENT.

See Figures 3 to 6 inclusive.

Below the "F" deck from forward aft, compartments between main bulkheads are—Fore Peak, Nos. 1, 2, and 3 Cargc Holds, Fresh and Distilled Water Tanks, Cofferdam, Fuel Oil Tanks, No. 1 Boiler Room with wing fuel tanks at sides, Auxiliary Engine Room with wing fuel and settling tanks at sides, No. 2 Boiler Room with wing fuel tanks at sides, Main Engine Room, Refrigerating Machinery Room with Fresh Water Tanks at sides, Shaft Alleys and No. 5 Cargo Hold, Fresh Water Tank, After Peak.

On the "F" deck compartments are—Stores, Chain Locker, 'Tween Decks Nos. 1, 2, and 3 Cargo Holds, Fresh and Distilled Water Tanks, Cofferdam, Fuel Oil Tanks, No. 1 Boiler Room with wing fuel tanks at sides, Generator and Switchboard flat of Auxiliary Engine Room with wing fuel and settling tanks at sides, No. 2 Boiler Room with wing fuel tanks at sides, Operating Platform and Workshop in Main Engine Room, Refrigerated Cargo, No. 5 Cargo Hold, No. 6 Cargo Hold, Stores.

On the "E" deck compartments are—Stores; Chain Locker; 'Tween Decks No. 1 Cargo Hold; Crew's Quarters and No. 2 Trunked Hatch; Cabin Baggage Room, No. 3 Trunked Hatch,

Swimming Pool and Mail Room; Boiler Hatch and Crew's Quarters; Storerooms, Workshop and Messrooms; Boiler Hatch, Cooks' Quarters and Messrooms; Engine Hatch, Cooks' Quarters, Third Class and Crew's Galley; Refrigerated Stores, No. 4 Trunked Hatch, and Third Class Dining Room; Third Class Staterooms and No. 5 Trunked Hatch; Third Class Staterooms and No. 6 Trunked Hatch; Steering Gear.

On the "D" deck compartments are—Stores; Crew's Quarters and No. 1 Trunked Hatch; Crew's Quarters, No. 2 Trunked Hatch, Registered Mail; Natatorium, Gymnasium, No. 3 Trunked Hatch, Cabin Staterooms; Cabin Staterooms, Boiler Hatch; Stores Working Passage, Auxiliary Machinery Hatch, Tourist Staterooms; Tourist Staterooms, Boiler Hatch; Tourist Staterooms, Engine Hatch; Tourist Staterooms, No. 4 Trunked Hatch; Tourist Staterooms, No. 5 Trunked Hatch; Tourist and Convertible Staterooms, No. 6 Trunked Hatch; Auxiliary Steering Gear.

On the "C" deck compartments are—Boatswain's Stores and Paint Locker; Crew's Quarters, No. 1 Trunked Hatch; Crew's Quarters and Hospital, No. 2 Trunked Hatch; Cabin Staterooms, No. 3 Trunked Hatch; Cabin Entrance Foyer and Purser's Office, Novelty Shop and Cigar Store; Cabin Dining Salon; Galley and Bake Shop; Tourist Dining Salon; Tourist Entrance Foyer and Staterooms. Aft on "C" deck are Emergency Generator Room, Third Class Lounge and Deck Spaces.

On the "B" deck forward are the Hawse Pipes, Windlass, Winches and Resistor Room, Nos. 1 and 2 Hatches. Compartments between fire bulkheads are—Cabin Staterooms, No. 3 Trunked Hatch and Foyer; Cabin Staterooms, Boiler Hatch and Dome over Cabin Dining Salon; Cabin Staterooms, Boiler Hatch and Engine Hatch; Cabin Staterooms, Dome over Tourist Dining Salon, No. 4 Trunked Hatch; Tourist Lounge and Smoking Rooms.

On "A" deck are No. 3 Hatch, Winches and Cabin Deck Space; Cabin Staterooms and Foyers; aft is the Tourist Game Deck.

Cabin Class Public Rooms, Promenade, and Game Deck occupy the Promenade Deck. From forward aft are the Palm Court, Entrance Lobbies and Pantries, Grand Salon, Writing Room and Library, Smoking Lounge, Pantries, Veranda Cafe.

The Boat Deck space is occupied by Officers' Quarters, the dome over the Grand Salon and Children's Play Room. Boats carried on Welin-MacLachlan davits leave a clear promenade on the Boat Deck. On the Sun Deck are the Bridge, Wheel House, Chartroom, and Cabin Game Deck.

Access through watertight bulkheads is provided by watertight doors, all below the " C" deck being motor operated with control switches on the bridge and on each side of the door. Manual operating gear extending above " C" deck is fitted to each door in addition to the motor operated gear. Fifteen horizontal sliding doors 6 feet 3 inches high by 2 feet 6 inches wide are located on " E" deck; six vertical sliding doors 4 feet 6 inches high by 2 feet 6 inches wide provide access between machinery spaces and into the shaft alleys. The motor operated watertight doors can be opened or closed from the bridge singly or together, alarm gongs connected with each sounding an alarm while the gear is operating.

HULL EQUIPMENT.

The windlass, located forward on " B" deck, is of the horizontal spur-geared type, with two wildcats, two gypsy heads, two slipping clutches and five hand-operated jaw clutches, on a common bed-plate for the windlass and motors. The motors are each of 90 horsepower at 400 R.P.M. One motor can be used on both wildcats when heaving short or under moderate load. When heaving in both anchors simultaneously under heavy load conditions, the center clutch is disengaged, and each motor drives its wildcat independently of the other. Anchors, cables, tow-lines, hawsers, and warps were supplied to meet the requirements of the Classification Society and the owners and include

3 Bower Anchors, stockless, 20,500 pounds each.
1 Stream Anchor, stockless, 7805 pounds.
330 Fathoms 3 7/16-inch stud link chain cable.

Aft, on " C" deck, are two warping capstans, with 26-inch diameter heads, driven through worm and spur-gearing by motors of 75 horsepower each at 600 R.P.M. An additional warping capstan of the same capacity but with motor mounted on the same bedplate is also located aft on the starboard side of " C" deck.

The main steering gear is of the four-cylinder, double-ram, hydro-electric type. The arrangement consists of 20-inch diameter cast steel cylinders on each side parallel to the centerline and forward of the rudder stock. Each pair of cylinders is yoked and bolted together at the center. The forged steel rams, turned and polished, are provided with upper and lower pins for the double link connections to the rudder yoke. The links are of forged steel and bushed at the ends. The gear is operated by duplicate horizontal motor-driven pumping units, one of which is a standy-by. Each pumping unit consists of a Hele-Shaw variable-stroke rotary cylinder high-pressure pump, direct driven by a motor of 90 horsepower at 1000 R.P.M. Steering is controlled from the wheel house by a single hydraulic telemotor system, the after telemotor connected through links to a follow-up pilot valve which controls the stroke of the main pump. By-pass valves are provided for by-passing oil when operating the auxiliary steering gears. A mechanical connection from the steering gear to the steering column on the docking bridge is provided, and operates through a clutch at the trick wheel at the forward end of the steering gear.

An auxiliary steering gear, of the motor-driven, quadrant type, driven through spur and bevel-gearing by a 25-horsepower motor is installed on the deck above the main steering gear. The cast steel quadrant is mounted on a 12-inch diameter extension of the rudder stock.

Cargo winches are of the motor-driven, single-drum, single-geared, high-speed type. No. 1 Hatch is served by four 3-ton winches and four 3-ton booms mounted on king-posts. No. 2 Hatch is served by three 3-ton winches and three 3-ton booms and one 20-ton boom on the foremast. No. 3 Hatch is served by three 3-ton winches and three 3-ton booms also on the foremast. Nos. 4 and 5 Hatches are each served by three 3-ton winches and three 3-ton booms on the mainmast. No. 6 Hatch is served by four 3-ton winches and four 3-ton booms mounted on king-posts. Booms are tubular steel and the masts of steel built up.

Cabin baggage and mail are handled through No. 3 Hatch, Tourist and Third Class baggage are handled through No. 6 Hatch.

The refrigerated cargo holds, served through No. 4 Hatch, have a capacity of 47,000 cubic feet and are divided into four compartments, with the lower end of the hatch trunk used as a handling room and emergency cargo hold. Ship's cold storage, on "E" deck, has a capacity of 16,000 cubic feet, divided into rooms for meats, fruits, vegetables, fish, butter, eggs and milk, ice cream and poultry. Insulation is of cork slabs, waterproof sheathing paper, and tongued and grooved spruce, except at the ship's side, where mineral wool is used instead of cork slabs. Holds and cold storage rooms are cooled by brine coils on the side and, where necessary, overhead. The compartments are independent of each other and can be cooled to the temperature suitable for the contents of each compartment.

The refrigerating plant, located between the shafts, next aft of the engine room, consists of three 2-cylinder, vertical, single acting, motor driven CO_2 Compressors, 5-inch diameter, 6-inch stroke, each having a refrigerating capacity of 37.6 tons per 24 hours at plus 5 degrees F. suction temperature and 86 degrees F. gas temperature; together with three CO_2 condensers and three evaporators; two main brine circulating pumps, one a stand-by, one smaller brine circulating pump for ship's service boxes; and two cooling water circulating pumps, one a stand-by. The plant is arranged so that one compressor can be used on dining salon cooling while one or both the other compressors operate on the refrigerating load.

The Cabin and Tourist Dining Salons each have two air conditioners cooled by CO_2 coils for summer conditions and two banks of steam heating coils for winter conditions.

A number of electric refrigerators are installed in pantries and service bars.

A very considerable quantity of insulation has been fitted under decks, on uptakes and casings, and on fire bulkheads.

EMERGENCY EQUIPMENT.

Sixteen lifeboats, 30 feet by 10 feet 6 inches by 4 feet 6 inches, 84 persons each, are carried on Welin-MacLachlan gravity type davits on the Boat Deck. Four identical lifeboats stowed in tiers of two boats each, aft on "A" deck, are handled by Steward

Mechanical davits. These boats are of copper bearing steel. See Figure 7.

Two motor lifeboats, 30 feet by 8 feet 8 inches, 424 cubic feet, with gasoline engine power for 8 knots equipped and loaded, and gasoline capacity for 200 miles, are carried aft on the Promenade Deck, and are equipped with radio transmitting and receiving sets and hinge masts for the radio antenna.

Twenty-one buoyant floats, 20 persons each, are stowed on the Sun, "A" and "B" decks, in locations suitable for quick handling in case of emergency. Solid cork life preservers, 1700 adult size, 167 children's size, 12 solid cork ring life buoys complete with hand lines and water lights, and 12 solid cork ring life buoys with hand lines are supplied for passengers and crew.

A "Rich" smoke detecting system is installed in cargo holds, dry stores, refrigerated cargo spaces, baggage rooms, mail rooms, linen storage rooms, boatswain's stores, lamp room, and paint room. The illuminating light in the smoke detecting cabinet in the pilot house is concealed and does not interfere with navigation at night.

The fire-extinguishing apparatus consists of a steam smothering system controlled through manifolds for the cargo holds, dry stores, refrigerated cargo, baggage rooms, mail rooms, and linen rooms, and a "Lux" CO_2 gas system for boatswain's stores, lamp room, and paint room. Manifolds with distribution valves permit discharge of gas into the affected compartments.

Portable hand fire extinguishers are scattered around the ship in suitable locations and a "Foamite" foam fire-extinguishing system is installed with hose outlets and hose at each fuel oil filling station, in each fireroom, and in the forward end of the engine room.

A fire main is fitted throughout the ship with hydrants and hose reels so located that any part of the ship can be reached with one length of hose.

Steel fire bulkheads, extending across the ship, faced with "Tucowul" on both sides, are fitted about 90 feet apart in passenger quarters.

A 40-zone mercurial thermostat fire alarm system is installed with annunciators, manual alarm boxes, alarm gong, and duplicate storage batteries, as required by the United States Steamboat Inspectors.

FIG. 7.—LIFE-SAVING EQUIPMENT OF THE MOST MODERN TYPE FEATURES THE "MANHATTAN."

FIG. 8.—GENERAL VIEW OF NAVIGATING EQUIPMENT IN THE PILOT HOUSE.

Two separate collision and alarm bell systems are provided, one for officers and crew and the other for passengers' quarters.

NAVIGATION EQUIPMENT.

All navigational equipment complies with U. S. Navy Specifications. The magnetic compass equipment consists of steering compass installed at the steering wheel in the pilot house, a standard compass at the trick wheel on top the pilot house, and a binnacle and compass on the docking bridge.

A complete gyro-compass system is installed; the master gyro installed in the gyro-room on "E" deck, with six repeaters, one in the wheel house, one on the gyro-pilot, one in the radio direction finder, one in the captain's office, one in the pelorus on the port wing of the bridge, and one on the docking bridge. A gyro-pilot is installed in the wheel house for automatic steering under control of the gyro-compass system. The gyro-pilot is of the double-unit type, with the steering unit in the pilot house and the control unit in the steering gear room. Course recorder and rudder angle recorders are installed in the chart room. Rudder angle indicators are mounted in the pilot house and control platform of the engine room. See Figure 8.

A shaft-revolution indicator system is installed with transmitters on the main gears, indicators in the pilot house, chief engineer's office, and indicators and a comparator on the control platform of the engine room.

A radio direction finder is installed in the pilot house and a fathometer in the chart room.

The engine order docking and steering telegraphs are the latest alternating-current type. The electrical engine-order telegraph system consists of two double-face, double-engine-order telegraph transmitters and receivers, of the pedestal type, located on either side of the pilot house, together with single-face, single-engine-order telegraph indicators with reply, of the bulkhead type, at each main throttle valve. The electrical docking and steering telegraph system consists of two single-face docking and steering telegraph transmitters and receivers, of the pedestal type, one in the pilot house and one on the docking bridge. An electrical fireroom telegraph system of the same make and type is installed, with

transmitter in the engine room near the gage board and indicators in each fireroom. A reserve mechanical engine-order telegraph system is installed, consisting of a double-face double-engine-order transmitter, pedestal type, in the pilot house, and two single-face, single-engine-order receivers in the engine room.

The radio installation is capable of holding uninterrupted day and night communication with either side of the Atlantic.

Two navigating telephone systems are installed. One system consists of five stations for communication between the pilot house, chief engineer's room, engine room, steering gear room, and docking bridge. The other system consists of three stations for communication between the pilot house, crow's nest, and forecastle.

INTERIOR ARCHITECTURE AND DECORATION.

The Cabin Class Public Rooms, Palm Court, Grand Salon, Library, Writing Room, Smoking Room, and Veranda Cafe are all located on the Promenade Deck, with convenient exits to the deck through vestibules between public rooms and doors at the after end of the Veranda Cafe. A complete promenade, enclosed with Kearfott sliding windows over the forward half, with large game deck space aft, thus surrounds the public rooms.

The Palm Court is decorated with Chinese scenes in subdued lacquer tones of rose, blue, and green. The furniture is Chinese Chippendale.

The Grand Salon is in the Georgian style with fluted Walnut pilasters with hand-carved caps and Walnut panels in the wall spaces. A stage is at the forward end and a concealed altar at the after end. Above the altar recess is the motion picture projection room.

The Library, on the starboard side, is panelled from deck to ceiling in square Old English Oak panels of Tudor period. The Writing Room, on the port side, is an adaptation of Heppelwhite style. The wall panels are of satinwood and Walnut with burl inlays.

The Lounge or Smoking Room is in a beam and column design incorporating Indian and Aztec decorative *motifs*. Four mural paintings by Lazzarini are set into the four sides of the ceiling

FIG. 9.—THE CABIN DINING SALON.

FIG. 10.—CABIN-CLASS FOYER.

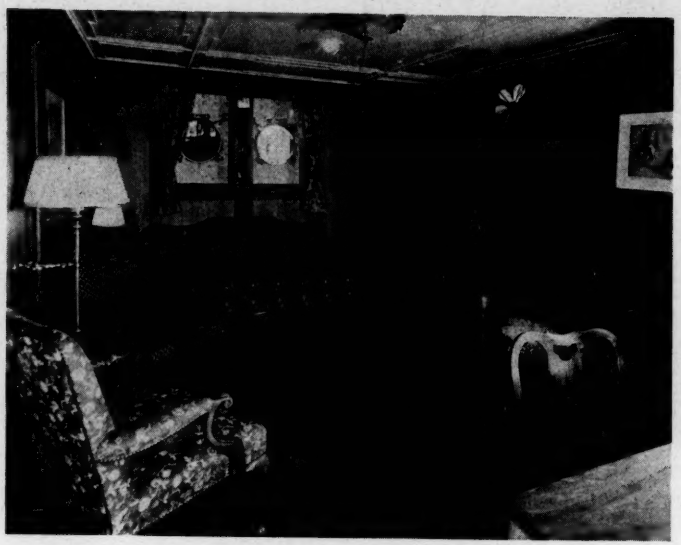


FIG. 11.—SUITE SITTING ROOM.

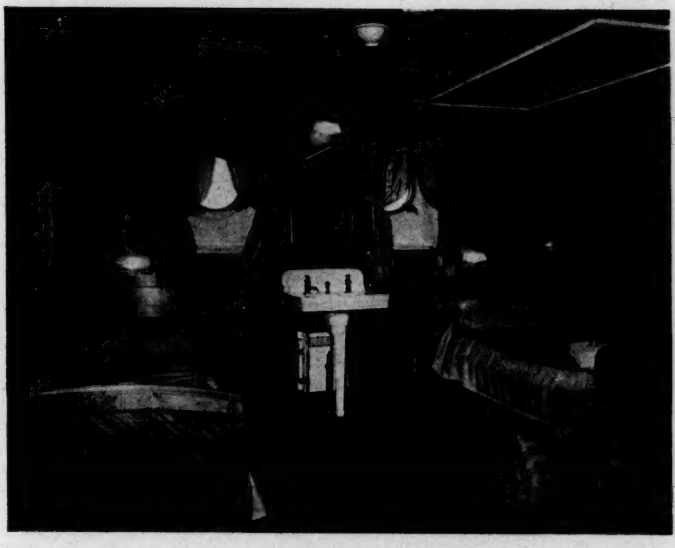


FIG. 12.—CABIN-CLASS STATEROOM.

FIG 13.—MAIN GALLEY ON "C" DECK.

well. These are Indian scenes—"The Buffalo Hunt," "The Scout," "The Water Hole," and "The War Dance." A wood-burning fireplace of generous dimensions extends across the forward end of the room.

The Veranda Cafe is in Venetian style with Walnut beams, twisted columns, and stencil painted plaster walls. An elliptical dance floor occupies the center of the room.

The Cabin Dining Salon carries out with considerable fidelity the Louis XVI period. Square columns support the ceiling well. The after side of the ceiling well is occupied by the orchestra balcony, the other three sides having the Lazzarini murals—"Feast of Bacchanal," "Diana and Her Attendants," and "Music—The Story of Lake Nemi." The dome over the ceiling well contains the Lazzarini mural, "The Judgment of Paris." A carved cornice surrounds the well. The lighting fixtures, supplied by the Sterling Bronze Company, are specially designed in Louis XVI manner and are fitted in clusters on the columns and sidewalls. Cove lighting behind recessed windows gives the illusion of daylight. See Figure 9.

The Tourist Lounge is of 18th Century English period with painted columns, beams, and wall panels. Walls are in cool pale green, and windows open on the tourist promenade space, "B" deck.

The Tourist Smoking Room is adapted from an early English tap room. Walls are in beam effect against rough cream-colored plaster.

The Tourist Dining Salon embodies an artist's conception of wooden shipbuilding. Massive columns and beams support the ceiling and ceiling well in which are placed the two Lazzarini murals "The Discoverers" and "The Pioneers."

Cabin Foyers, Passages and Staterooms are panelled with flameproof plywood panels faced with exotic wood veneers, with contrasting pilasters and moldings. Tourist staterooms and passages are in white with Mahogany doors. Third-class staterooms and passages are in white with Oak doors. All staterooms of all three classes have lavatories with hot and cold fresh water. Cabin staterooms have standing wooden beds, with concealed fold up auxiliary Pullman berths. Tourist staterooms have standing

metal beds with fold-up auxiliary Pullman berths. Third-class staterooms have standing metal beds with upper berths. Cork tiling for insulation and sound absorption is fitted on the deck under the carpeting in all Cabin staterooms. The majority of the Cabin staterooms have private baths. Twelve suites, eight on "A" deck and four on "B" deck, supply the demand for more luxurious accommodations. See Figures 10 to 12 inclusive.

HOTEL SERVICE.

The Main Galley is located on the "C" deck between the Cabin and Tourist Dining Salons and serves both. Including the pantries and shops at the side of the ship it occupies a space 80 feet wide by 105 feet long. Cooking equipment is in general electrical. The central section, fore and aft, contains the working kitchen of a battery of 12 electric ranges and 3 steam tables. Pantries, butcher shops, bake shops, pastry shop, vegetable rooms, dish-washing rooms, etc., flank the working kitchen on both sides. Mechanical appliances and labor-saving devices are too numerous to mention. See Figures 13 and 14.

The main ventilation systems are of the heated air supply type. The supply fans discharge through thermo-tanks and the air, warmed as necessary, is distributed by ducts to "Punkah-Louvre" terminals. The galleys have independent supply and exhaust systems. The exhaust system for the main galley has 100 per cent greater capacity than the supply system in order to draw in air from the dining salons and eliminate entrance of galley fumes. Ventilating ducts in cabin and tourist accommodations are concealed behind ceilings, only the louvres being visible. For "boosting" the temperature of staterooms to meet individual requirements or in severe cold weather, there are fitted radiators supplied with hot water from the ship's hot-water system. For heating public rooms, public toilets and officers' quarters on the boat deck a two-pipe heating system is installed to work at 45 pounds pressure. Radiators in public rooms are concealed behind metallic ornamental grilles.

Hot and cold fresh water is supplied to all lavatories, bathtubs, and showers. Cold salt water is supplied for flushing all water closets and urinals, and supplying all slop sinks. The water serv-

FIG. 14.—GALLEY COFFEE PANTRY, LOCATED ON "C" DECK.

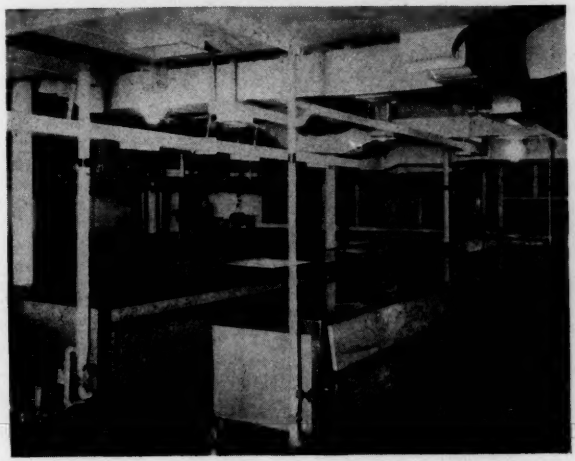


FIG. 14.—ELECTRIC RANGES AND STEAM TABLES.

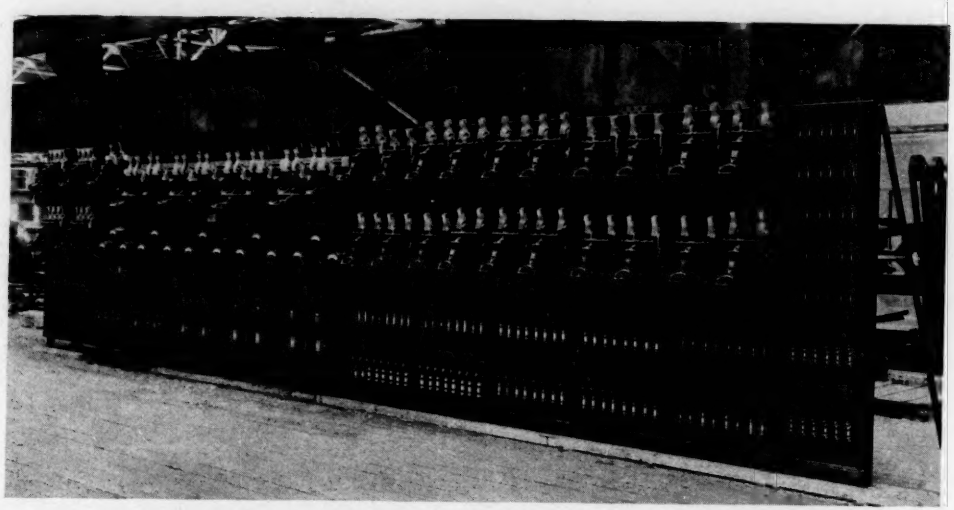


FIG. 15.—MAIN SWITCHBOARD DESIGNED AND BUILT BY THE SHIPYARD, AS IT APPEARED SET UP IN THE SHOP BEFORE INSTALLATION.

ices are pressure systems, arranged in sections so that one section may be cut out without interfering with the operation of the system. Soil lines drain overboard independent of all other piping. Drains and soil lines in each watertight compartment are independent of piping in adjacent compartments. Sanitary discharges from the " E" deck are disposed of overboard by seven duplex sewage ejectors.

Complete and extensive call bell systems are installed, calling stewards or stewardesses to any stateroom or public space. Cabin staterooms are fitted with desk type monophone telephones connected up with a 320-line single-position switchboard.

Each stateroom is equipped with an electric bracket fan.

Two passenger elevators for cabin passengers are installed abreast the main stairway forward and serve six stops from the " D" deck to the boat deck. An engineer's elevator serves four stops from the engine room to the boat deck. A stores elevator for the galley serves " C" deck and " E" deck.

ELECTRIC PLANT AND INSTALLATION.

Four 500 Kw., 3-wire, turbo generators are installed in the auxiliary engine room, protected from damage by collision by wing tanks. Current is supplied at 240 volts for all power purposes and at 120 volts for lighting. For emergency use a 75 Kw., 3-wire generator driven by a Diesel engine is installed aft on " C" deck, supplying current at 240 volts for the emergency bilge pump, watertight doors, and windlass, and through a compensator supplying current at 120 volts for emergency lighting, flood lights for lowering lifeboats, and for navigating lights and equipment. For instant use, in emergency, current is available from a 240-volt storage battery.

The distribution system is arranged so that auxiliary switchboards and panels are located in spaces where heavy loads occur. The main switchboard, 31 feet 3 inches in length, contains the meters, circuit breakers, and switches for the generators, auxiliary generators, and distribution of current to the auxiliary switchboards and panels, there being 9 auxiliary switchboards, 29 power panels, and 85 lighting panels installed in the locations best suited for convenient operations. See Figure 15.

PROPELLING MACHINERY.

The power plant of the *Manhattan* is a twin-screw single reduction geared-turbine installation with high pressure, high temperature boilers, designed for 30,000 shaft horsepower at 125 R.P.M. of the propellers.

BOILERS AND FIREROOMS.

Steam is supplied at 400 pounds 675 degrees F. by six Babcock & Wilcox express boilers, three abreast in each fireroom.

Total Boiler Water Heating Surface, square feet	63,000
Total Superheating Surface, square feet	15,042
Total Air Heating Surface, square feet	49,242
Total Furnace Volume, cubic feet	7,572

See Figure 16.

Each boiler is equipped with ten Babcock & Wilcox Cuyama burners, soot blowers, feed water regulators, drum and superheater safety valves, and smoke indicators. Tubular air heaters are installed in the uptakes immediately above the boilers.

Four motor-driven forced draft blowers, located in a separate blower room, supply air to each fireroom. Closed fireroom forced draft is used with an air pressure of approximately three inches at full power. The blowers draw air down from cowls on the fidley top through ducts in the stack casing and discharge above the operating space in front of the boilers. Air for combustion enters the air heaters at the back of the boilers at the upper level of the fireroom and from the second pass of the air heaters passes through ducts around and under the boilers into the double front around the burners.

There are two smoke pipes, each serving one fireroom.

Saturated steam at 125 pounds pressure for auxiliary purposes, heating systems, galley, tank coils, fuel oil heating, etc., is supplied by two high-pressure evaporators, one in each fireroom. Heating steam is supplied from the main boilers and under normal conditions "raw" fresh water is used for high-pressure evaporator feed. The arrangement protects the main boiler superheaters, separates the main boiler cycle from the saturated steam cycle, and supplies distilled make up feed when vapor from the evaporators is led to the second stage feed water heater.

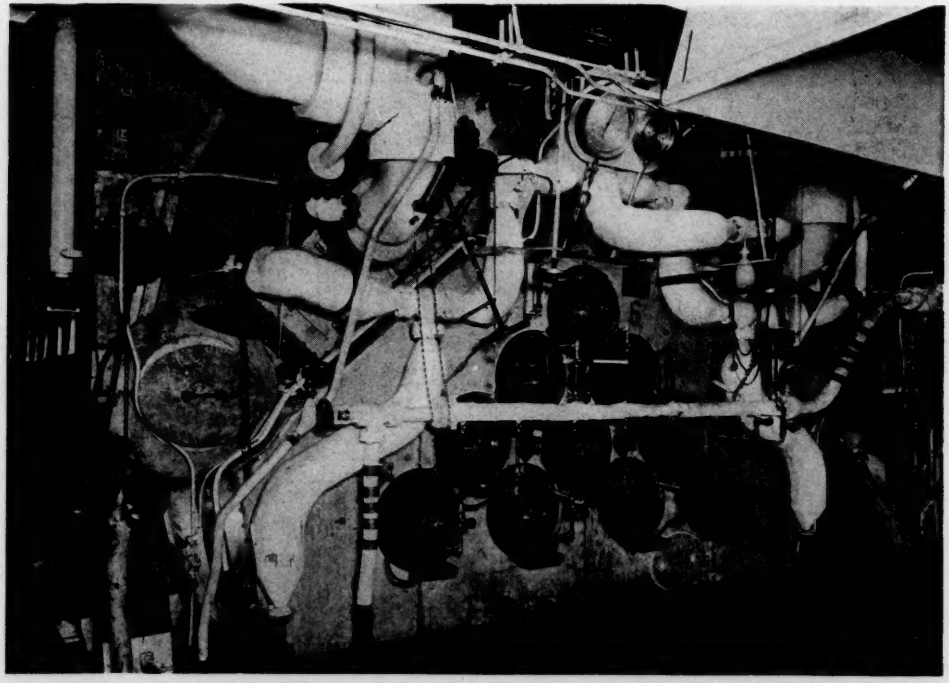


FIG. 16.—FRONT OF ONE OF THE BOILERS, SHOWING THE BURNER ARRANGEMENT.

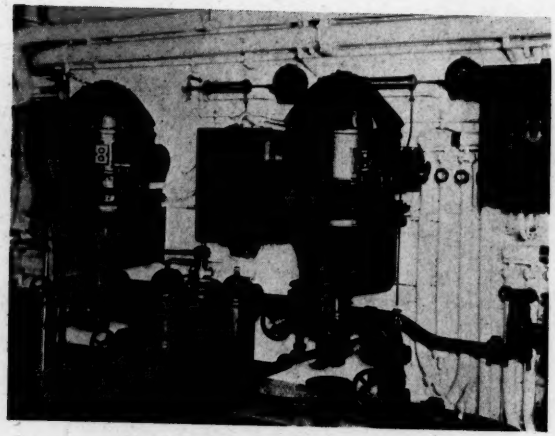


FIG. 17.—FUEL OIL SERVICE PUMPS.

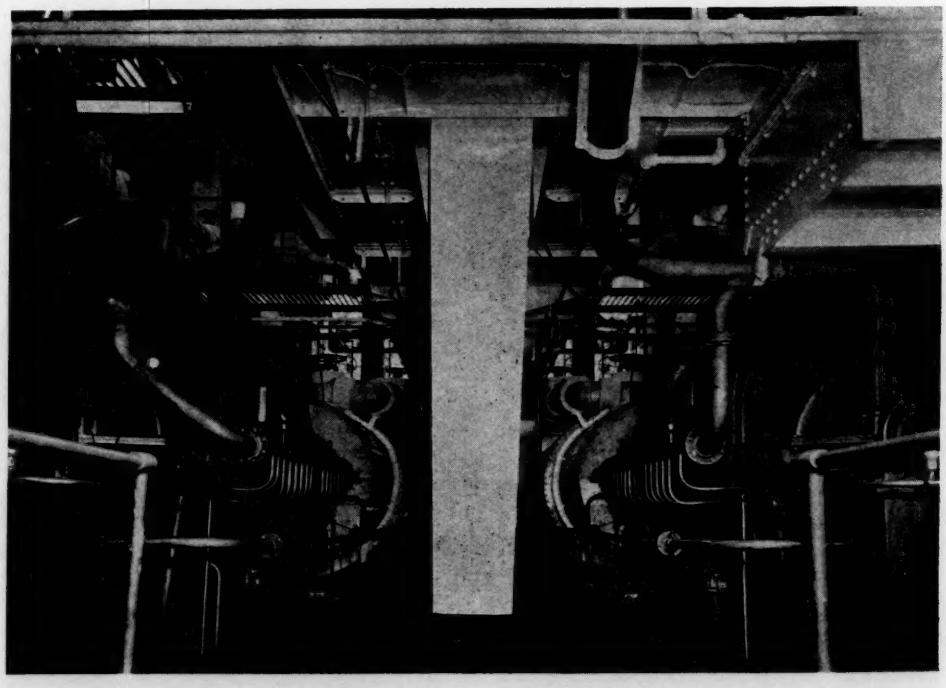


FIG. 18.—ENGINE ROOM, LOWER LEVEL.

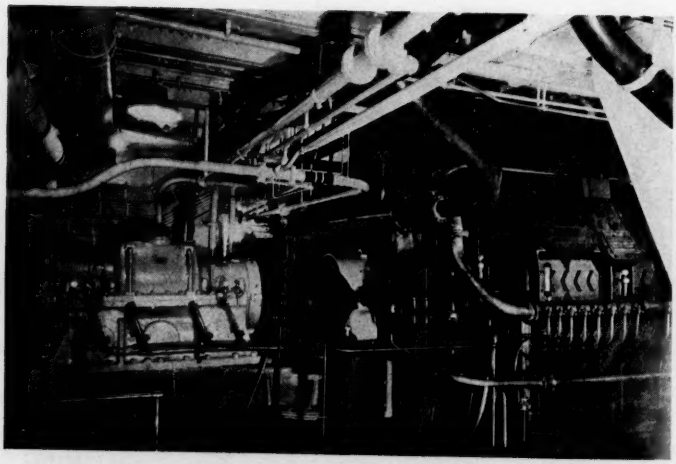


FIG. 19.—REDUCTION GEAR AND THRUST BEARING.

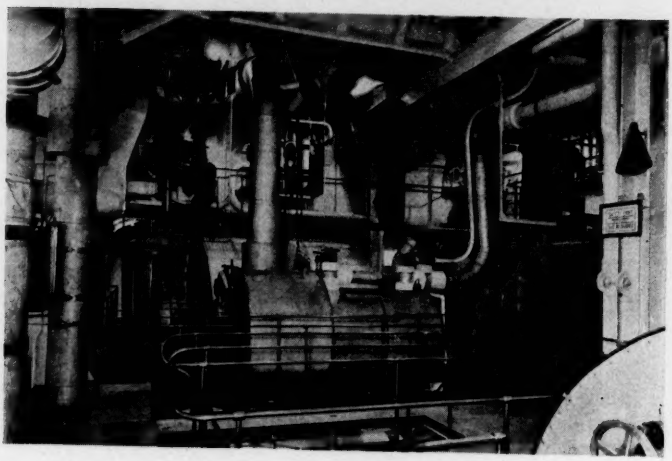


FIG. 19A.—STARBOARD PROPELLING UNIT.

A CO_2 indicating and recording instrument is installed in each fireroom.

Each fireroom is independent of the other. In each there are two 12-inch by 8-inch by 18-inch vertical simplex emergency feed pumps, two vertical fuel oil service pumps and one fuel oil transfer pump of the same type. Two fuel oil heaters with a section for drain sub-cooling are installed in each fireroom, one a stand-by. A small rotary gear pump is installed in No. 2 fireroom for starting up from cold. In No. 1 fireroom there is a 10-inch by 12-inch by 24-inch emergency fire and bilge pump and in No. 2 fireroom are two centrifugal self-priming bilge and ballast pumps. See Figure 17.

AUXILIARY ENGINE ROOM.

All auxiliary machinery associated with hotel service or in use in port is installed in the auxiliary engine room. On the upper level are the turbo-generators and switchboard. Below are the auxiliary condensers, circulating pumps, condensate pumps and port feed pump. On the lower level are also two centrifugal fire pumps, two centrifugal salt water sanitary and wash-deck pumps, two centrifugal fresh washing water pumps, two hot fresh water circulating pumps, two drinking water pumps, two evaporators, two distillers and two fresh water storage tank heaters.

ENGINE ROOMS.

The main turbines and gears are of simple, rugged design with conservative speeds, stresses and tooth pressures. See Figures 18, 19 and 19a.

Each set of turbines consists of three New York Ship-Parsons turbines in series, with separate H.P. Astern turbine, arranged around a single gear wheel. The L.P. Astern turbine is incorporated in the forward end of the L.P. Ahead turbine. The H.P. Astern turbine is in tandem with the I.P. Ahead turbine.

The H.P. Ahead turbines take steam through hand controlled nozzle valves and have an initial stage impulse wheel followed by end-tightened reaction blading. Impulse blading and nozzles are of corrosion-resisting steel. End tightened reaction blading is of manganese copper. Regular blading is of hard-drawn brass. The casing is of cast steel with a contact dummy at the forward

end. The thrust bearing is adjustable. The turbine design was based on full power with 85 per cent nozzle area open.

The I.P. Ahead turbine is fitted with reaction blading throughout. A radial dummy is fitted at the after end.

The L.P. turbine is of the balanced double-flow reaction type.

The H.P. Astern turbine consists of two impulse stages and exhausts into the L.P. Astern turbine, consisting of two impulse stages.

All turbine glands are of labyrinth design with a common sealing and leak-off system. The outside chambers of the glands are connected to a fan-suction leak-off condenser. Two gland leak-off systems are provided, one for each propelling unit.

Bleeder nozzles are arranged in the H.P. turbine at the wheel chamber and bleeder belt, in the I.P. receiver, and in the I.P. bleeder belt for supply of heating steam to the feed water heaters. The higher pressure points supply the second-stage heater and the I.P. bleeder supplies deficiency in auxiliary exhaust for the first stage heater.

The two independent main condensers are single-pass, scoop injection with auxiliary pump, of about 16,500 square feet cooling surface each, with 30 per cent nickel "Ambrac" tubes. Tubes are ¾ inch O.D. No. 16 B.W.G. packed at after ends, rolled at inlet ends. Shells are of steel.

The reduction gears, manufactured by the Falk Corporation, are of three pinion, single wheel, double-helical design. The H.P. pinion is at the top, I.P. pinion on inboard side, L.P. pinion on outboard side. The pinions are each of 13.429 inches pitch diameter and the gear wheels of 161.429 inches pitch diameter. Total width of gear face is 62 inches. Motor driven turning gear is attached to the after end of the I.P. pinion and arranged so that when the gear is engaged a red warning light shows on the main gage board. See Figure 20.

The main thrust bearings are each mounted in independent steel housings with a journal at each end. The thrust collar is 41½ inches in diameter.

Propellers are solid, of manganese bronze.

For each propelling unit there are installed two vertical, centrifugal, motor-driven condensate pumps, two vertical motor-driven lubricating oil pumps and one double two-stage air ejector.

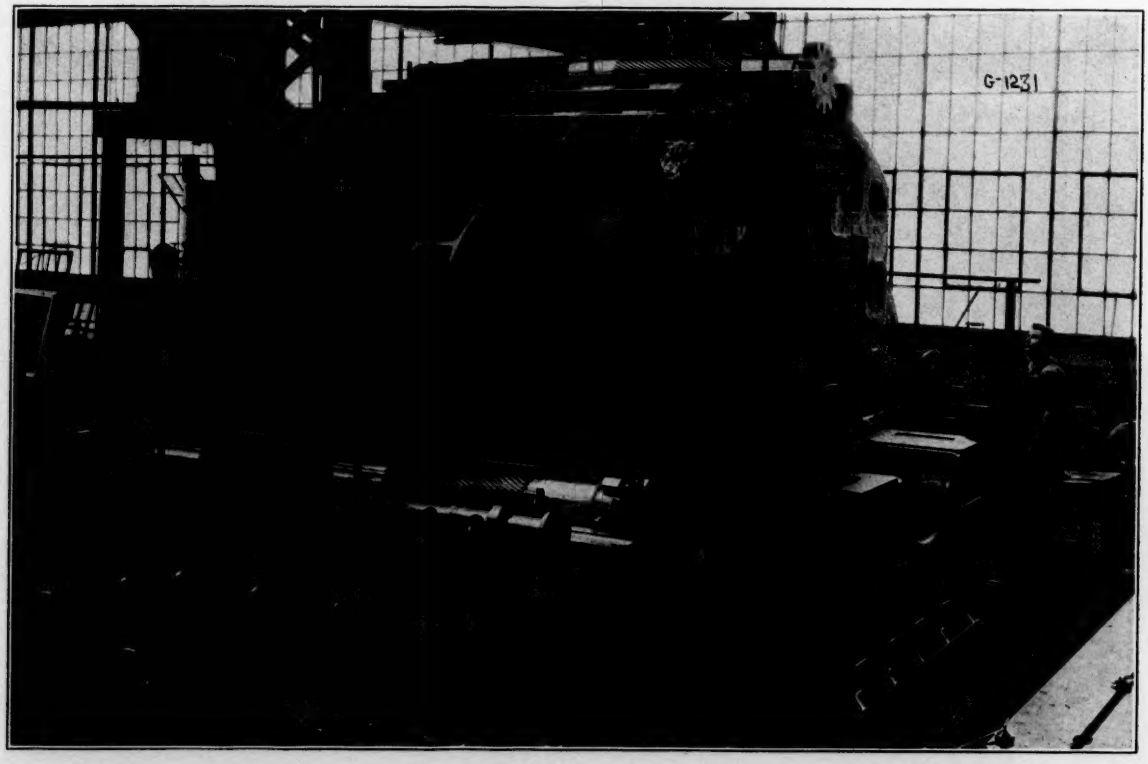


FIG. 20.—MAIN REDUCTION GEAR.

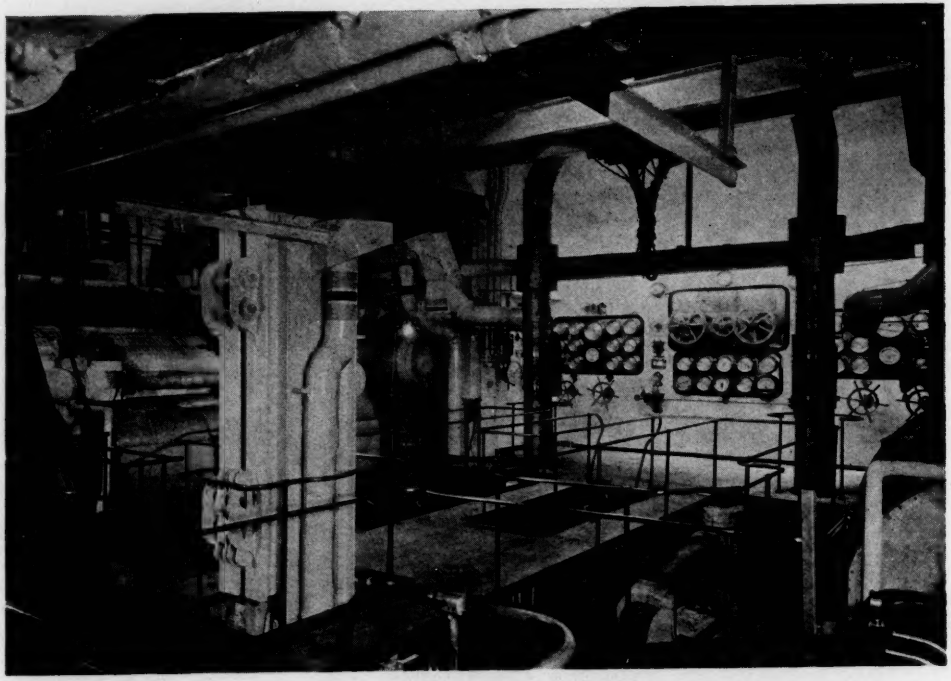


FIG. 21.—ENGINE ROOM, SHOWING OPERATING PLATFORM.

Two turbine-driven four-stage main feed pumps and one similar auxiliary feed pump are installed at the forward end of the engine room.

The feed system is of the immediately available storage closed type, the feed tank floating on the discharge from the condensate pumps to the feed pump.

Two lines of main steam piping are installed, one port and one starboard, both leading from No. 1 fireroom to the bulkhead stops at the operating platform and cross-connected so that steam from any or all boilers may be supplied to the engine room through either or both lines. Expansion is taken up by bends, there being no expansion joints in the system.

The operating platform is arranged to provide simple and easy handling and maneuvering, gages for each propelling unit being located immediately above each throttle valve wheel with a central gage board for other gages and instruments. Telegraphs are at the sides. In addition to gages and instruments there are indicator lights for the forced draft blowers, lubricating oil system, ventilation fans, as well as the rudder indicator, feed tank water-level indicator and the emergency hand control for main steam bulkhead valves. See Figure 21.

The gage board bulkhead is placed in front of the bulkhead and throttle valves, only the handwheels showing through. The space behind the board, containing the valves, steam pipes, strainers, etc., is connected to the exhaust ventilating system. Excessive heat at the operating platform is thus eliminated.

Two workshops are provided, one on the starboard side of the operating platform and containing the heavier machine tools, the other on "E" deck near the auxiliary engine room hatch.

FIG. 21.—ENGINE ROOM, SHOWING OPERATING PLATFORM.

TRIAL TRIP.

En route to Rockland, Maine, on Sunday, July 24th, power was steadied at 30,000 S.H.P. and a four-hour trial conducted to demonstrate the guaranteed water rate of the main turbines, check instruments, and rehearse observers for the official fuel consumption trial. The guaranteed water rate of 7.75 pounds per S.H.P. hour was easily met; the measured water rate corrected to standard conditions, 375 pounds, 650 degrees F. being 7.65 pounds per S.H.P. hour.

The standardization trial was held on July 25th. Three runs each at 10, 15, 18 and 20 knots and five runs at full power were made over the Rockland measured mile. Averaged data is as follows:

STANDARDIZATION.

Knots	*R.P.M.*	*S.H.P.*
10.388	54.79	2629
15.427	82.16	8862
18.199	99.75	16,074
20.231	111.23	22,634
22.696	128.91	36,620

Following the standardization trial, and after reaching deep water, the contract 8-hour fuel consumption trial was conducted. Averaged data is as follows:

R.P.M.	125.20
S.H.P.	33,085
Generator, Kw.	700
Generator, H.P.	1118
Total horsepower	34,203
Fuel per hour corrected to 18,500 B.T.U. per pound, pounds	20,450
Fuel per total H.P. per hour, pound	0.597
Steam pressure at boilers, pounds per square inch	401
Temperature at superheater outlet, degrees F.	636
CO_2 per cent	14.5
Fuel oil at burners, pounds per square inch	185
Temperature, degrees F.	183
Baumé at 60 degrees F.	13.3
B.T.U. per pound	18,500
Uptake temperature, degrees F.	328
Air pressure in fireroom, inches water	3
Air at burners, degrees F.	322
Steam at H.P. chest, pounds per square inch gage	385
Steam at H.P. first stage, pounds per square inch gage	263
Steam at I.P. chest, pounds per square inch gage	83
Steam at L.P. chest, vacuum, inches mercury	3
Main condenser, vacuum, inches mercury	29.13
Temperature, main injection, degrees F.	65
Temperature, overboard discharge, degrees F.	71
First stage feed water heater, feed water in, degrees F.	98
First stage feed water heater, feed water out, degrees F.	234
Second stage feed water heater, feed water out, degrees F.	319

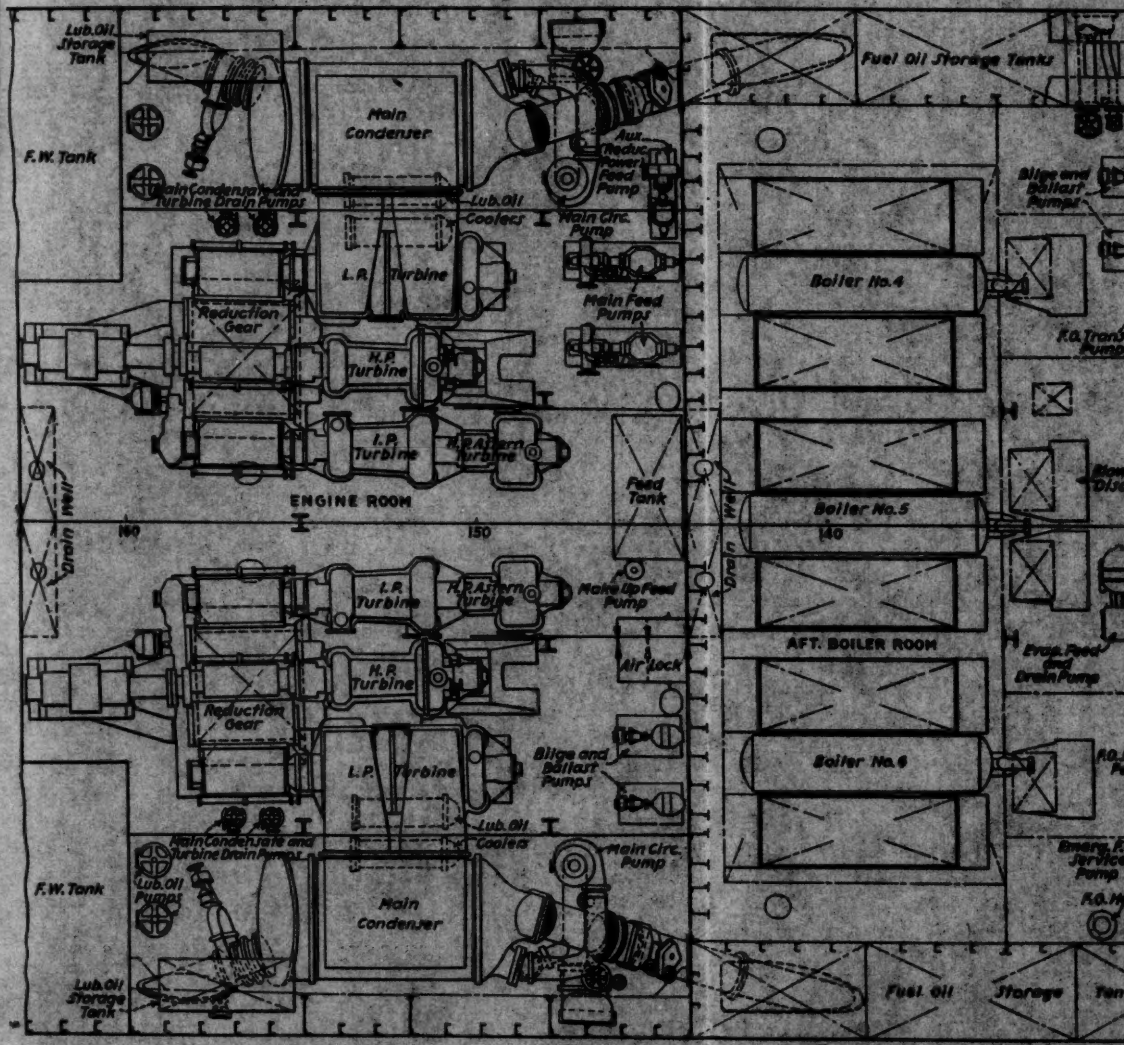
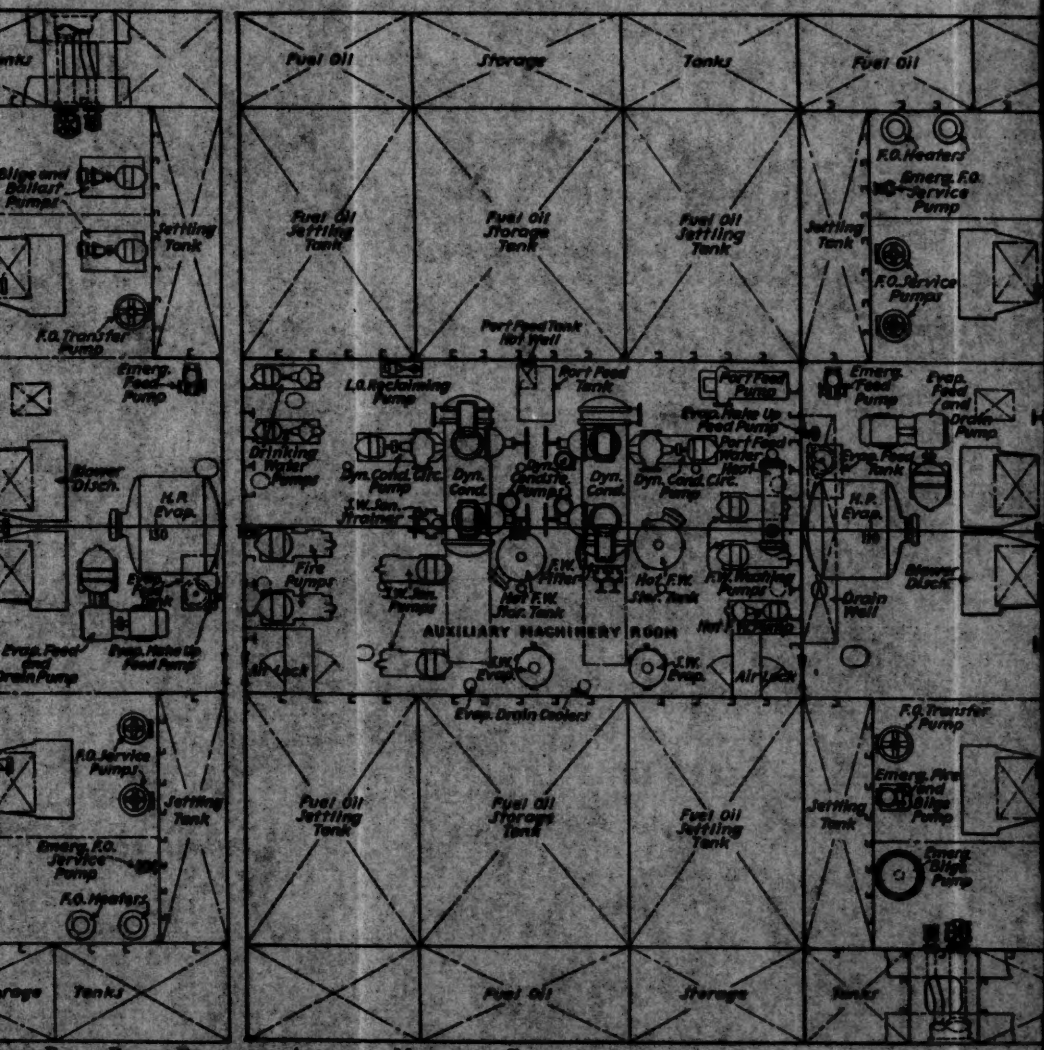


FIG. 22.—PLAN OF ENGINE ROOM,



INE ROOM, BOILER ROOMS AND AUXILIARY MACHINERY ROOM.

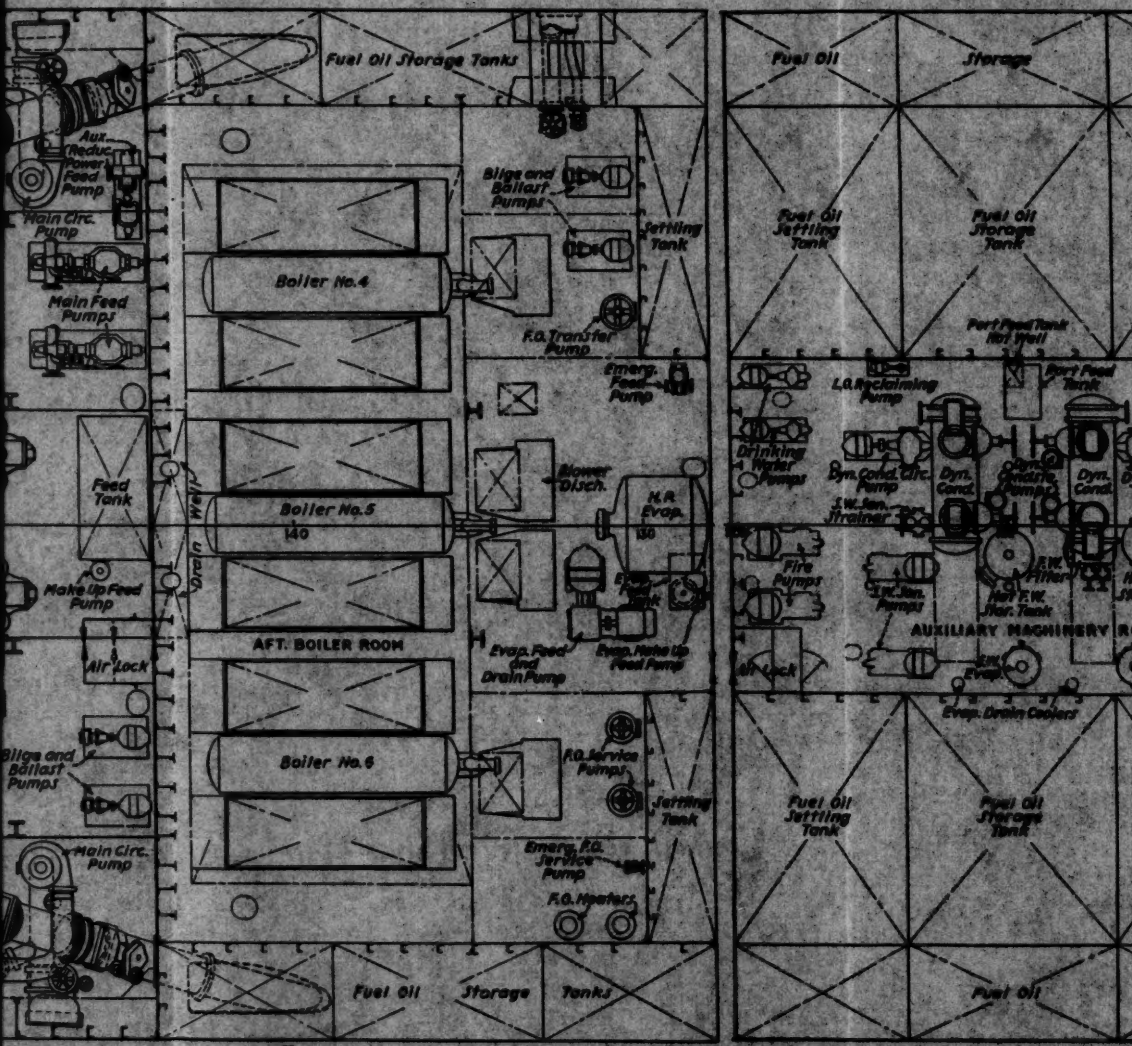
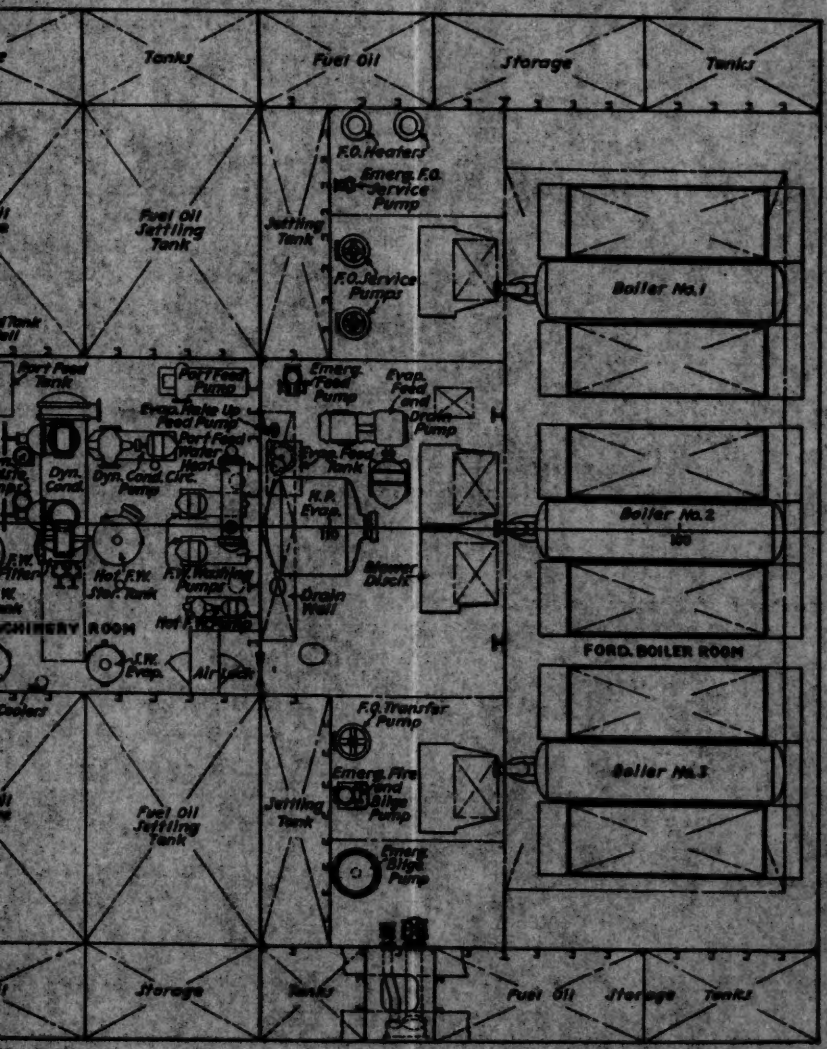


FIG. 22.—PLAN OF ENGINE ROOM, BOILER ROOMS AND AUXILIARY MACHINERY ROOM.


Tanks
Fuel Oil
Storage
Tanks
F.O. Heaters
Emerg. F.O. Service Pump
Fuel Oil Settling Tank
Settling Tank
F.O. Service Pumps
Boiler No. 1
Port Feed Tank
Port Feed Pump
Emerg. Feed Pump
Evap. Feed and Drain Pump
Evap. Make Up Feed Pump
Port Feed Heater
Dyn. Cond.
Dyn. Cond. Circ. Pump
Evap. Feed Tk.
R.P. Evap.
Boiler No. 2
Blower Ducts
Hot F.W. Stor. Tank
Drain Well
FORD. BOILER ROOM
F.O. Transfer Pump
Emerg. Fire and Bilge Pump
Boiler No. 3
Emerg. Bilge Pump
Storage
Tanks
Fuel Oil
Storage
Tanks

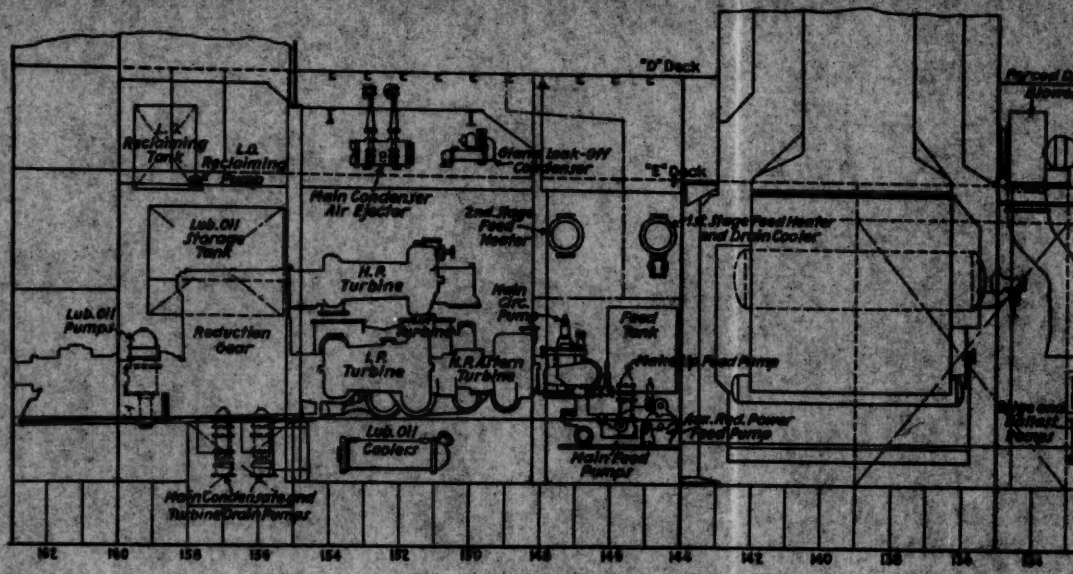


Fig. 23.—Elevation of Engine Room, Bo

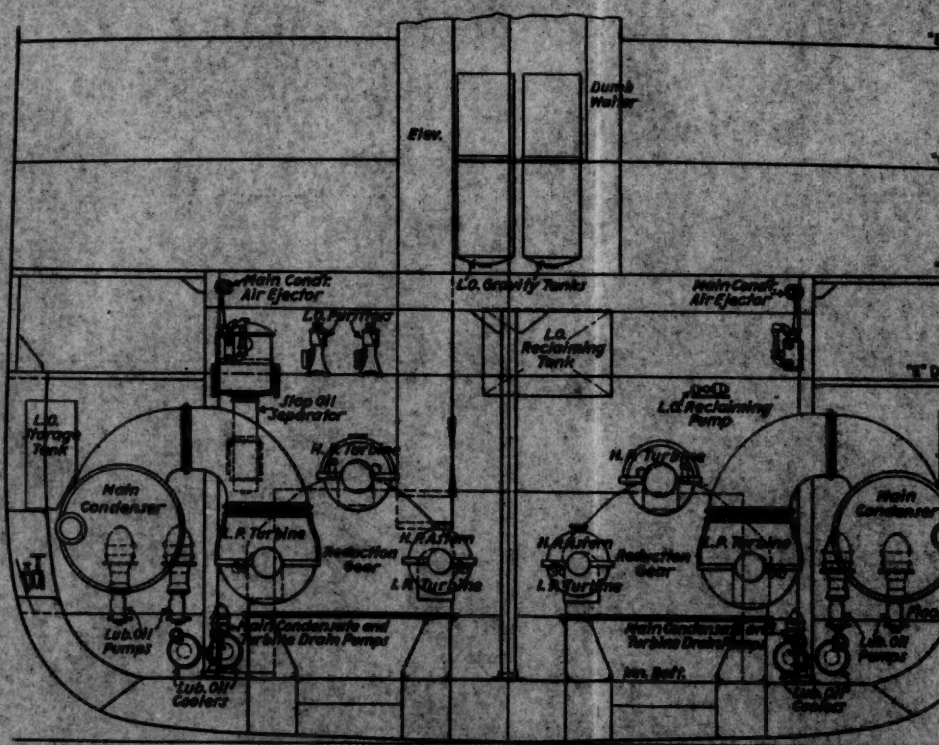
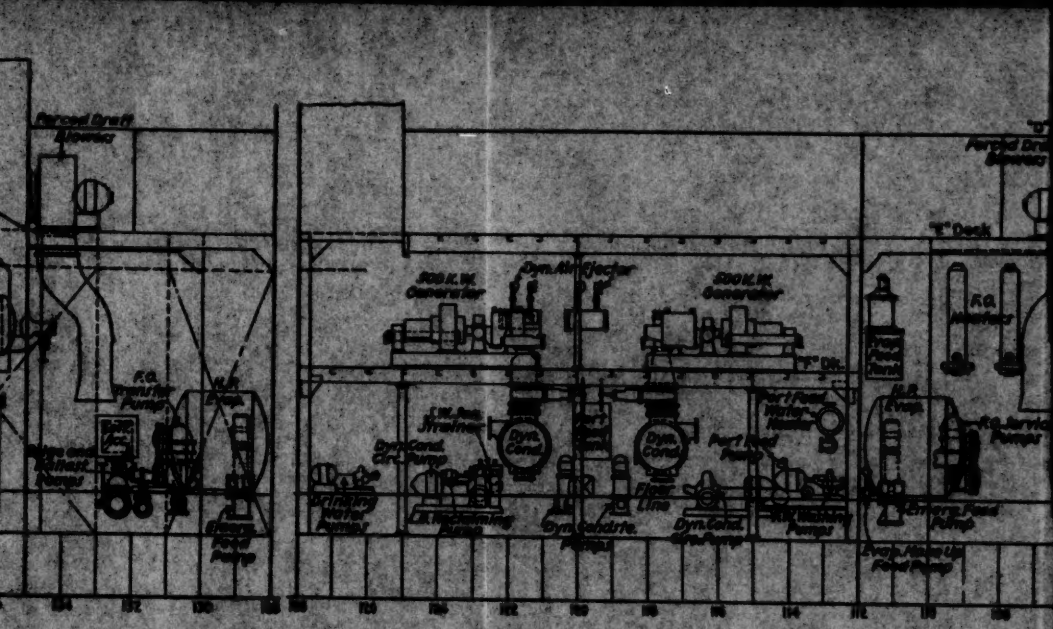


Fig. 24.—Section Through Engine Room at Frame 155, Looking Aft.



Room, Boiler Rooms and Auxiliary Machinery, Port Side Looking Outboard.

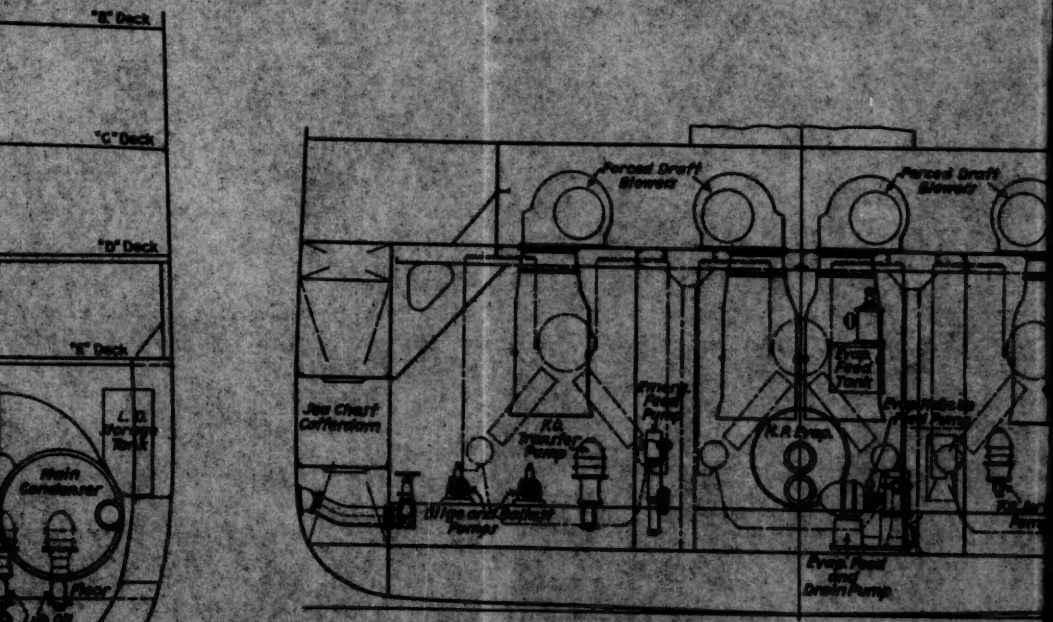


Fig. 25.—Section Through After Boiler Room at Frame 128, L

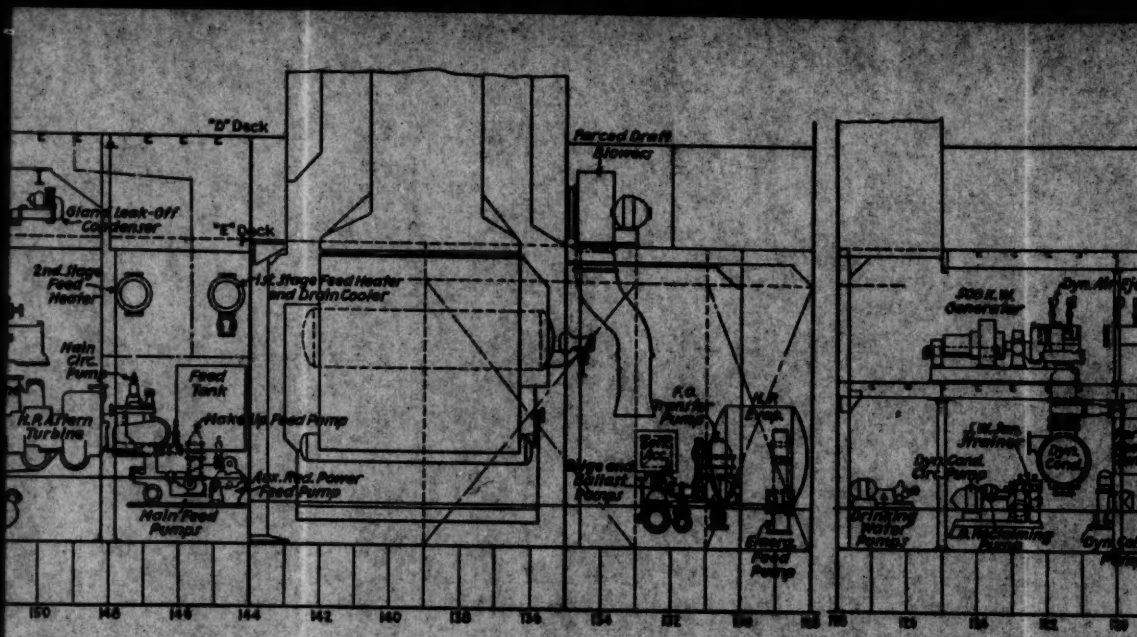
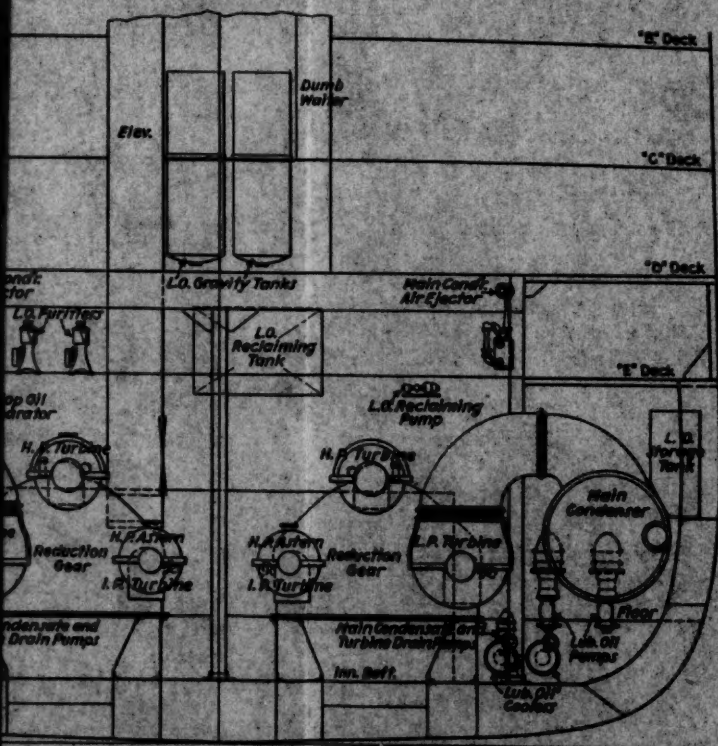


FIG. 23.—ELEVATION OF ENGINE ROOM, BOILER ROOMS AND AUXILIARY MACHINERY, PORT SIDE LO



SECTION THROUGH ENGINE ROOM AT FRAME 155, LOOKING AFT.

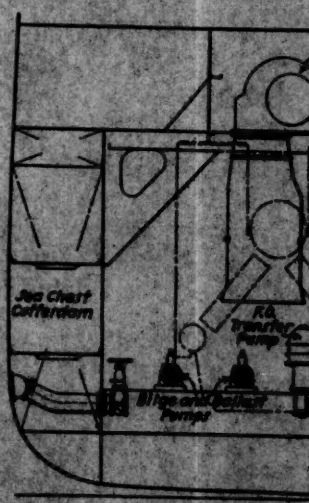
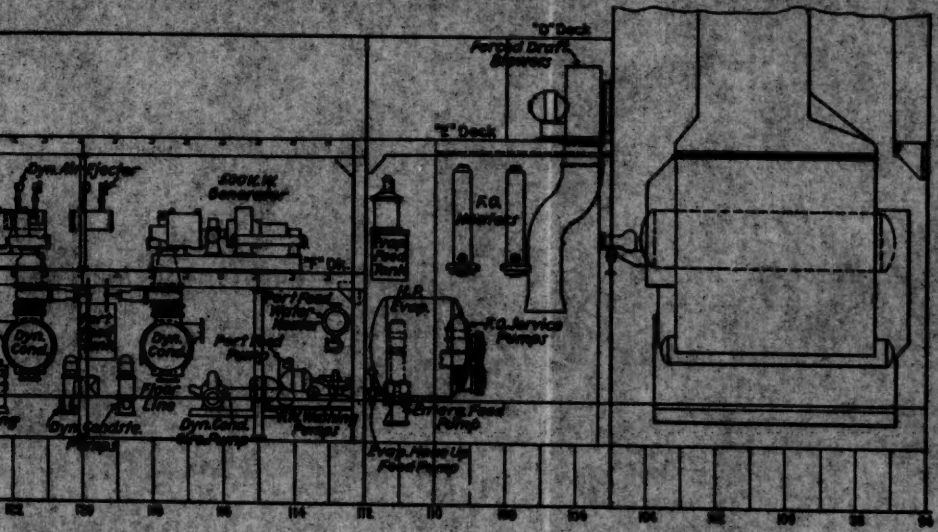
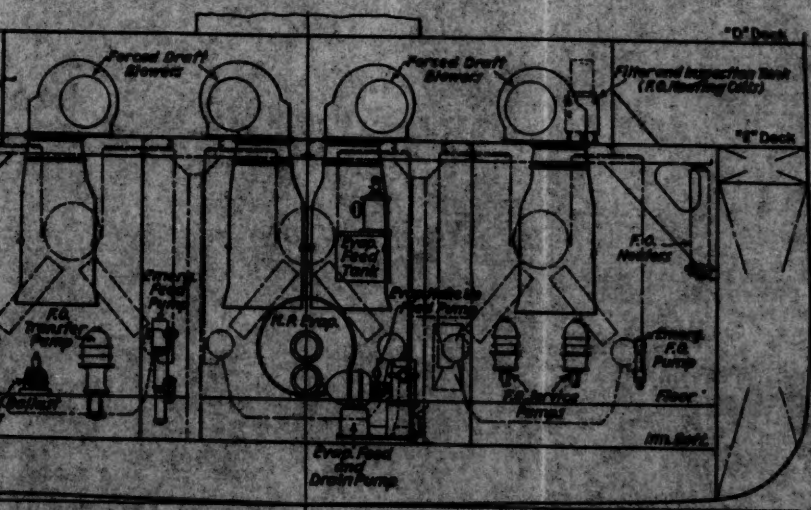


FIG. 24.—SECTION



ART SIDE LOOKING OUTBOARD.



—SECTION THROUGH AFTER BOILER ROOM AT FRAME 124, LOOKING FORWARD.

This descriptive article would be incomplete without appreciative acknowledgment of the sincerity of purpose displayed upon many occasions by officers and ex-officers of the Navy in the Navy Department and U. S. Shipping Board. The disturbed economic status of shipping during the building period gave opportunity for intrigue and malicious reports attempting to discredit the builder's design. The responsible officers of the Navy and Shipping Board, charged with the duty of passing upon calculations and plans, refused to be swayed from sound principles and the whole truth truthfully told. To these, on behalf of the members of the staff of the New York Shipbuilding Company, the writer expresses acknowledgment and appreciation.

TABLE OF MACHINERY.

See Figures 22 to 25 inclusive.

Main Turbines.

Parsons turbines—Falk reduction gears—twin screw.

Turbine chest steam—375 pounds gage—650 degrees F. Temp.—208 degrees superheat.

Vacuum in condenser—29 inches.

Ahead nozzles—16 open + controlled groups, 2, 4, 8 = 30 total.

Full Power (2 shafts) (specified)—30,000 S.H.P.

Propeller R.P.M.—125—Turbine R.P.M. = 1500.

85 per cent of nozzles to be sufficient for full power = 25 nozzles.

Maximum Power (designed)—33,000 S.H.P.

Propeller R.P.M. = 131—Turbine R.P.M.—1570.

If more than 25 nozzles are required, open bypass valve around 1st group of reaction blading.

Governor Setting on Turbines—1660 R.P.M.

Overspeed Test—1725 R.P.M.

Astern Power (specified)—65 per cent of ahead full power.

Bleeder Valves—Nonreturn valves on all turbine bleeder connections are provided with adjustable spring to equalize flow of steam from port and starboard turbine.

Emergency Lubricating Oil Low Level Alarm and Shut-Off.

Winter's alarm bell rings—when level of lubricating oil in gravity tank drops 2 feet.

Emergency stop valve shuts off turbine—when level of lubricating oil in gravity tank drops 7 feet 6 inches.

6 Boilers—(B. & W. Type)—(Closed Fireroom).

Boiler steam (at supht. outlet) 400 pounds—225 degrees superheat—675 degrees temperature.

Safety valve setting (superheater valve 401 pounds)—(drum valve 409 pounds).

Pressure drop through superheaters at designed power—6 pounds to 8 pounds.

	1 Boiler	Total
Boiler heating surface (evaporating)	10,500	63,000
Superheaters	2625	15,750
Air heaters	8207	49,242
Boiler efficiency, per cent		84.7
Uptake temperature at designed power not to exceed, degrees		350

60 Oil Burners (Cuyama Type).

OIL PRESSURE CAPACITIES, POUNDS PER HOUR.

Size Tip	150 *Pounds Pressure*	200 *Pounds Pressure*	250 *Pounds Pressure*	300 *Pounds Pressure*
No. 4008	357	414	462	508
No. 4408	293	339	375	408
No. 5520	167	180	183	184

4 Fuel Oil Heaters (Davis).

Capacity each—12,000 pounds per hour of heavy gravity fuel oils (700 seconds 120 degrees S.F.) from 60 degrees to 300 degrees F.

Steam pressure—90 pounds.

Drains from heater not to exceed 150 degrees F.

4 Fuel Oil Service Pumps (Quimby).

Vertical motor driven, screw type.

Capacity 34 G.P.M.—300 pounds discharge pressure—handling light or heavy gravity fuel oil 375 to 700 Sec. S.F.

Westinghouse motor—17 H.P. 450/1350 R.P.M.

2 Emergency Fuel Oil Service Pumps (Worthington).
6½ × 3½ × 8 V.S. steam 375 pounds saturated 10 pounds B.P.
Capacity 22 G.P.M.—300 pounds discharge pressure—45 feet P.S.

2 Fuel Oil Transfer Pumps (Quimby).
Vertical motor driven, screw type.
Capacity—300 G.P.M. 75 pounds discharge pressure—handling light or heavy gravity fuel oil 375 to 700 Sec. S.F.
Westinghouse motor—30 H.P. 400/1200 R.P.M.

8 Forced Draft Blowers (Sturtevant).
Horizontal motor driven.
Capacity 20,300 cubic feet/min. at 4 inches static pressure.
Westinghouse motor—20 B.H.P.—450/750 R.P.M.

2—Main Condensers (N. Y. S. Co. manufacture).
Single pass—16,500 square feet cooling surface.
Vacuum 28.8 inches to 29 inches—sea temperature 70 degrees bar. 30 inches.
Circulating water—scoop injection at speeds above about 10 knots; use circulating pumps below 10 knots.

2—Main Circulating Pumps (Worthington).
These pumps used for low speed and maneuvering only.
Turbine driven, vertical, centrifugal screw type.
Steam conditions: 375 pounds—225 degrees S. 10 pounds B.P.
Capacity:
Condenser duty: 18,000 G.P.M. 25-foot head.
165 H.P.—1090 R.P.M.
Emergency bilge duty 8,000 G.P.M. 35-foot head.
100 H.P.—820 R.P.M.

4—Main Condensate and Turbine Drain Pumps (Worthington).
Motor driven, vertical, centrifugal, single stage.
Motor: 10 H.P. 1350 R.P.M.
Capacity:
Main condensate pumps 300 G.P.M.—60-foot head.
Turbine drain pump—15 G.P.M.—18-foot head.

4—Air Ejectors for Main Condensers (Westinghouse).

Guarantee—1 first and 1 second stage ejector to remove 45 pounds dry air per hour with entrained moisture from its condenser.

Steam—250 pounds gage—225 degrees superheat—controlled by throttling valve.

Cooling water = condensate from main condenser.

Temperature control valve recirculates condensate if outlet temperature from air ejectors exceeds 150 degrees F. This control valve adjustable for 140 to 180 degrees.

Evacuating capacity of air ejectors sufficient to obtain 20 inches vacuum in 8½ minutes and 29 inches vacuum in 15 minutes or less with all jets in operation.

Propellers.

Diameter, feet and inches	19.0
Pitch, feet and inches	20.0
Pa/Da	.383
Height of centerline of wheel above base line, feet and inches	10.6
Submergence of propeller:	
24 feet 0 inches draft aft, feet and inches	4.0
29 feet 0 inches draft aft, feet and inches	9.0

2—Main Feed Pumps (DeLaval).

Turbine driven, horizontal, centrifugal, 4-stage.

CAPACITIES.

	G.P.M.	*Pounds Per Hour*	*Pounds Feed Discharge Pressure*	*R.P.M.*
Maximum capacity	825	412,000	509	2700
Normal capacity	630	315,000	509	2580
Reduced capacities	315	158,000	509	2490
	150	75,000	509	2470

Capacity based on 4-foot head on suction and 130 degrees F. water temperature.

Steam—380 pounds gage—225 degrees superheat—10 pounds back pressure.

Nozzles—2 open nozzles + 4 single nozzles each hand controlled. For best economy nozzles to be shut off to give minimum pressure drop across governor valve and minimum number of nozzles in operation.

Turbine—350 H.P. at 2700 R.P.M.

2—Auxiliary Feed Pumps (DeLaval).

Turbine driven, horizontal, centrifugal, 4 stage.

CAPACITIES.

	G.P.M.	*Pounds Per Hour*	*Pounds Feed Discharge Pressure*	*R.P.M.*
Maximum capacity	150	75,000	509	3500
Normal capacity....	100	50,000	509	3340
Reduced capacities	60	30,000	509	3290
	10	5,000	509	3260

Capacity based on 4-foot head on suction and 130 degrees F. water temperature.

Steam—380 pounds gage—225 degrees superheat—10 pounds back pressure.

Nozzles—3 single nozzles, each hand controlled.

Turbine—100 B.H.P.—3500 R.P.M.

Auxiliary feed pump in engine room can be used for low speeds up to about 12 knots.

Auxiliary feed pump in generator room for port use.

2—Emergency Boiler Feed Pumps (Worthington)—12 × 8 × 18 V.S.

Capacity—156 G.P.M. (78,000 pounds per hour)—500 pounds discharge pressure at 60-foot P.S.

Steam conditions—375 pounds, saturated 10 pounds B.P.

1—1st Stage Feed Water Heater and Drain Cooler (Schutte Koerting).

Horizontal—(4) pass in heater (1) pass in cooler—straight tube.

Capacity—335,000 pounds per hour from 110 to 225 degrees with 8 pounds shell pressure.

Temperature of drains not to exceed 150 degrees F.

Test pressure on shell 200 pounds.

Feed pressure drop not to exceed 14 pounds for heater and drain cooler.

Shell relief valve set at 30 pounds.

1—2nd Stage Feed Water Heater (Schutte Koerting).

Horizontal—four pass—straight tube.

Capacity—335,000 pounds per hour from 225 to 300 degrees F. with 70 pounds shell pressure.

Test pressure on shell 200 pounds.

Feed pressure drop not to exceed 11 pounds.

Shell relief valve set at 150 pounds.

1—Port Feed Water Heater and Drain Cooler (Schutte Koerting).

Horizontal (4) pass in heater and (1) pass in cooler—straight tube.

Capacity 75,000 pounds per hour from 100 to 225 degrees with 8 pounds shell pressure.

Temperature of drains not to exceed 150 degrees F.

Test pressure on shell 200 pounds.

Feed pressure drop not to exceed—5 pounds for heater and drain cooler.

Shell relief valve set at—30 pounds.

4—Lubricating Oil Coolers (Griscom Russell).

Horizontal, single pass.

Cooling water from main condenser scoop or circulating pump.

Capacity each—500 G.P.M.—D.T.E. heavy medium from 105 to 95 degrees with 80 degrees sea water.

Capacity two coolers in series—500 G.P.M. D.T.E. heavy medium from 120 to 95 degrees with 80 degrees sea water.

4—Lubricating Oil Pumps (Quimby).
Motor driven—vertical, gear in head, screw type.
Capacity each 600 G.P.M. (D.T.E. extra heavy) 35 pounds gage discharge pressure.
Westinghouse motor 30 H.P.—400/1200 R.P.M.

2—Lubricating Oil Reclaiming Pumps (Worthington).
1 pump for main L.O. system and (1) pump for 500 Kw. sets.
Motor driven—horizontal rotary gear type.
Capacity each—10 G.P.M.—10 pounds discharge pressure.
Motor—¾ H.P.—870 R.P.M.

2—Lubricating Oil Separators (DeLaval).
Motor driven—open type.
Capacity—300 G.P.H. lubricating oil at 160 degrees.
Motor—2 H.P.—1750 R.P.M.

Lubricating Oil Tanks.
2—Gravity tanks—1500 gallons each to overflow.
2—Storage tanks—3000 gallons each.
1—Reclaiming tank 3000 gallons.
2—Sump tanks 1500 gallons each.

4—Bilge and Ballast Pumps (Nash).
Motor driven, horizontal centrifugal, self priming.
Capacity—475 G.P.M. 75-foot discharge head.
Vacuum pump 25 G.P.M.—8-inch vacuum—capable of priming longest suction line in 6 minutes.
Westinghouse motor—15 H.P. 1310/1750 R.P.M. 230 volts D.C.

1—Emergency Fire and Bilge Pump (Worthington).
10 × 12 × 24 inches V. S.
Steam—375 pounds—10 pounds B.P.
Capacity—525 G.P.M.—125 pounds discharge pressure—90 feet P.S.

1—Emergency Bilge Pump (Worthington).
Motor driven vertical centrifugal submerged type.
Capacity—500 G.P.M.—55 feet total head.

Westinghouse motor—15 H.P. 1300 R.P.M. 230 volts, D.C.

Priming pump mounted on main pump shaft with jets installed at deck.

2—Fire Pumps—(Worthington).

Motor driven horizontal centrifugal, 2 stage.

Capacity—500 G.P.M.—125 pounds discharge pressure.

Westinghouse motor—60 H.P.—1150/1750 R.P.M. 230 volts D.C.

2—Salt Water Sanitary Pumps (Worthington).

Motor driven, horizontal, centrifugal, 2 stage.

Capacity—400 G.P.M. 75 pounds discharge pressure—83 pounds shut off pressure.

Westinghouse motor—30 H.P. 1150/1750 R.P.M., 230 volts D.C.

2—Fresh Water Washing Pumps (Worthington).

Motor driven, horizontal centrifugal, two stage.

Capacity—300 G.P.M.—60 pounds discharge pressure 70 pounds shut off pressure.

Westinghouse motor—20 H.P.—1750 R.P.M.

2—Hot Fresh Water Circulating Pumps (Worthington).

Motor driven, horizontal, centrifugal, single stage.

Capacity—160 G.P.M. 60 pounds discharge pressure, 45 pounds suction pressure.

Westinghouse motor 5 H.P. 1150 R.P.M. with 15 per cent overspeed and 10 per cent speed reduction.

2—Ice Water Circulating Pumps (Worthington).

Motor driven, horizontal, centrifugal, two-stage.

Capacity 30 G.P.M. 50 pounds discharge pressure—51 pounds shut off pressure.

Westinghouse motor 5 H.P. 1750/1375 R.P.M.—230 volts D.C.

2—Drinking Water Pumps (Worthington).

Motor driven, horizontal, centrifugal, two-stage.

Capacity 60 G.P.M. 75 pounds discharge pressure—85 pounds shut off pressure.

Westinghouse motor—15 H.P.—1750 R.P.M. 230 volts D.C.

2—Swimming Pool Filling and Drainage Pumps (Worthington).
Motor driven, vertical, "Freflo" type, single stage.
Capacity, 250 G.P.M. 31 feet static head, 16 pounds shut off pressure.
Westinghouse motor, 5 H.P. 1150 R.P.M.

1—Swimming Pool Circulating Pump (Worthington).
Motor driven, horizontal, centrifugal, single stage.
Capacity 50 G.P.M. 10 pounds discharge pressure, 11 pounds shut off pressure.
Westinghouse motor 1 H.P. 1150 R.P.M. 230 volts D.C.

2—Fresh Water Heaters (Griscom Russell).
Capacity of each = 15 tons per hour (67 G.P.M.) (50 degrees to 140 degrees) with 90 pounds steam = 475 gallons storage capacity.
Temperature control on water outlet set at 140 degrees F. (range 100-160 degrees).
Temperature control on drains set at 150 degrees F. (range 140-200 degrees).

1—Salt Water Heater (Griscom Russell).
Capacity—50 G.P.M. (50 degrees to 110 degrees F.) with 90 pounds steam.
Temperature control for swimming pool set at 70 degrees F. (range 60-100 degrees).
Temperature control on drains set at 150 degrees F. (range 140-200 degrees).

Make Up Feed for Main Feed System.
Aerated feed water to be avoided as much as possible by taking make up feed via feed water heater or condensers. The methods of supplying make up feed as follows:
(1)—H.P. evaporator vapor to 2nd stage F.W. heater and to port feed water heater.
(2)—Distilled water feed tank connections to "dynamo condensers."
(3)—Make up feed pump connection from distilled water tank.
(4)—Emergency boiler feed pump from reserve feed tanks or from H.P. evaporator feed tanks.

Make Up Feed Pump for Main Feed System (Worthington).
Motor driven, vertical centrifugal type.
Capacity—100 G.P.M. at 20 pounds discharge pressure.
Westinghouse motor—4 H.P. 1150 R.P.M. 230 volts D.C. constant speed.
Duty—make up feed (emergency use) or for filling distilled water tanks from surplus in feed system.

2—Fuel Oil Inspection Filter Tanks (Davis Engineering Company).
Capacity 10,000 pounds of water per hour.

2—Grease Extractors for H. P. Evaporator Feed (Andale Engineering Co.).
Capacity each 25,000 pounds per hour—180 pounds feed discharge pressure.
Water temperature—150 to 180 degrees F.
Pressure drop—about 1 pound clean.

2—Fans for Gland Leak Off Exhausters—(Sturtevant Mfg. Company).
Capacity each 250 C.F.M. against 4 inches static head.
Motor—1 H.P. 2560 R.P.M.—50 per cent speed reduction.

4—Filters for Drinking Water System (Filtrine Manufacturing Company)
Each having a capacity of 30 G.P.M.
2 units working in parallel, 2 units as standby.
Pressure drop through filters about 3 pounds clean.

1—Oily Ballast Separator—(Condenser Service and Engineering Company).
Rated capacity—100 tons per hour—maximum 125 tons per hour when pumping ballast water from tanks previously containing oil not exceeding 12 degree Baumé.
Overboard discharge not to contain over .01 of one per cent of oil to comply with law.
Oil retrieved to be available as fuel.
Back pressure valve set at—15 pounds.

1—Filter for Fresh Wash Water Pump Discharge—(Griscom Russell).
Capacity—300 G.P.M.
Pressure drop through filter about—0 pounds clean.

7—Sewage Ejectors (Ansonia).
Ejectors capable of a maximum lift of 25 feet.
Air pressure for operating—100 pounds.
Quantity of air to be provided—150 cubic feet per minute total.

2—Air Compressors (Chicago Pneumatic).
Motor driven vertical duplex, double acting—two stage.
Capacity each—535 cubic feet per minute free air—100 pounds pressure.
Westinghouse motor—100 H.P.—450 R.P.M.—230 volts D. C.
Automatic starting at 90 pounds and stopping at 100 pounds pressure.

4—Wash Down Injectors (Schutte Koerting).
Capacity each 20 G.P.M.—50 pounds discharge pressure 135 to 165 degrees F.
Working steam pressure—90 pounds gage.

REDUCING VALVES, GOVERNOR VALVES, BACK PRESSURE VALVES, ETC.

Reducing Valves to Heating System (Foster).
1—3 inches normal capacity 0 to 7000 pounds per hour—maximum 10,000 pounds per hour.
1—3½ inches normal capacity 0 to 9500 pounds per hour—maximum 14,000 pounds per hour.
Pressure reduced from 125 pounds to 45 pounds.

Reducing Valves for Galley (Foster).
1—1½ inches normal capacity 0 to 1200 pounds per hour—maximum 2000 pounds per hour.
Pressure reduced from 125 pounds to 45 pounds.
2—¾ inches—capacity 0 to 300 pounds per hour.
Pressure reduced from 45 pounds to 10 pounds.

Reducing Valves for Laundry (Foster).
1—1¼ inches—capacity 0 to 1400 pounds.
Pressure reduced from 125 pounds to 80 pounds.

Reducing Valves for Deaerator (Foster).
2—4 inches—capacity 0 to 1800 pounds per hour.
Pressure reduced from auxiliary exhaust to 2 pounds.
These valves must be tight seating for dead end service.

Reducing Valves for Emergency Connection from Boilers to 125 Pounds Line (Leslie).

2—2 inches—normal capacity 500 to 11,000 pounds per hour—limited maximum capacity 15,000 pounds per hour.

Pressure reduced from 409 pounds to 125 pounds and adjustable for outlet presures (90 to 125 pounds).

Reducing Valve for 125 Pounds Line Connection to Port Feed Water Heater (Leslie).

1—2½ inches limited maximum capacity 6500 pounds per hour.

Pressure reduced from 125 pounds to 10 pounds (adjustable for 10 to 20 pounds).

Back Pressure Valves (Auxiliary Exhaust to Main Condensers) (Atwood Morrill).

2—6 inches—set at 15 pounds pressure.

Capacity 20,000 pounds per hour with not exceeding 2 pounds over pressure.

Atmospheric Exhaust Valve for Auxiliary Exhaust to Atmosphere (Atwood Morrill).

2—5 inches set at 25 pounds pressure.

Capacity 15,000 pounds per hour with not exceeding 6 pounds over pressure.

Back Pressure Valve (Auxiliary Exhaust to Dynamo Condenser) (Atwood Morrill).

2—4 inches set at 15 pounds pressure.

Capacity 7500 pounds per hour with not exceeding 2 pounds over pressure.

Atmospheric Relief Valves for 500 Kw. Sets.

4—6 inches valves open to atmosphere if vacuum fails in condenser.

Capacity 13,500 pounds per hour. Must be tight under high vacuum.

Governor Valves for Emergency Feed Pumps (Leslie).

2—1½ inches, adjustable for 50 to 100 pounds excess feed line pressure.

Governor Valve for Emergency Fire and Bilge Pump (Leslie).

1—1¼ inches, for 125 pounds constant discharge pressure.

Governor Valves for Make-Up Feed Pumps for H.P. Evaporator Plant (Leslie).

2—⅜ inches, for 35 pounds constant discharge pressure.

H. P. Turbine Exhaust Bleeder Valves (Atwood Morrill).

2—6 inches non-return—capacity 20,000 pounds per hour.
Outlet pressure—85 pounds gage—110 degrees superheat.
Designed W.P. = 125 pounds gage. Tight seating against 30 inches vacuum.

I. P. Turbine Bleeder Valves (Atwood Morrill).

2—8 inches non-return and stop valve—capacity 20,000 pounds per hour.
Outlet pressure—16 pounds gage—10 degrees superheat.
Designed W.P.—30 pounds gage. Tight seating against 30 inches vacuum.

4—Turbo Generator Sets (Westinghouse).

3 Wire—240/120 volts D.C.
Normal capacity—500 Kw.
Overload capacity—625 Kw. (2 hours).
Turbine: 6,000 R.P.M. Generator: 1200 R.P.M.
Steam pressure—390 pounds gage—207 degrees F. superheat—650 degrees temperature 28½ inches vacuum.
Lubricating oil = D.T.E. heavy medium or D.T.E. extra heavy.
L.O. cooler capacity = D.T.E. heavy medium cooled 115 degrees to 105 degrees with 85 degrees sea.

2—Dynamo Condensers (Westinghouse).

Two pass—2,000 square feet cooling surface.
Vacuum: 28.44 inches to 28.5 inches—sea temperature 80 degrees barometer 30 inches.
Circulating water—3600 G.P.M.
Pump discharges cross connected so that (1) pump can serve either or both condensers when desired.

2 Sets—Air Ejectors for Dynamo Condensers (Westinghouse).

Guarantee—1 first and 1 second stage ejector to remove 22 pounds dry air per hour with entrained moisture.
Steam—250 pounds gage—225 degrees F. superheat—controlled by throttling valve.

Cooling water—condensate from condenser.

Temperature control valve recirculates condensate if outlet temperature—150 degrees F.

Temperature control valve adjustable for (140-180).

3—Dynamo Condensate Pumps (Worthington).

Motor driven, vertical, centrifugal, single stage.

Westinghouse motor—3 H.P. 1750 R.P.M.—230 volts D.C.

Capacity—40 G.P.M. each—70 feet head.

2—Dynamo Condenser Circulating Pumps (Worthington).

Motor driven, horizontal, centrifugal, single stage, double suction.

Westinghouse motor—25 H.P. 850/1150 R.P.M. (variable speed).

Capacity each 3600 G.P.M. at 20 feet head.

1—Emergency Diesel Generator Set (Otto-Westinghouse).

Generator: (Westinghouse) 75 Kw. 3 wire D.C. 240/120 volts.

Diesel Engine: (Otto) airless injection—6 cylinder, 4 cycle —115 H.P.

Diesel engine to operate on 28 degrees to 34 degrees Baumé Diesel oil.

Starting air pressure—300 pounds.

ELECTRICAL LOAD.

Approximate normal loads on generating plant as follows, with two (2) generator sets operating:

At anchor	810 Kw.
At 10 knots	850 Kw.
At maximum speed	950 Kw.

When using air conditioning apparatus, windlass or other abnormal loads a third machine will be required.

2—Salt Water Evaporators and Distillers (Griscom Russell).

	SINGLE EFFECT (Each Evaporator)	DOUBLE EFFECT (2 Evaporators)
Rated capacity (tons per 24 hr.) (long tons)	25 tons	33 tons
Gallons per hour	280 gal.	370 gal.
Overload capacity	30 per cent	30 per cent

Purity of vapor.....	(not to exceed 2 gr. per gal.)	
Rated steam pressure in coils.....	90 pounds	90 pounds 1st—30 pounds 2nd
Rated shell pressure....	8 pounds	30 pounds 1st— 2 pounds 2nd
Shell relief valve setting	30 pounds	30 pounds
Shell water level.....	Float Control	Float Control
Shell concentration (maximum)	(5000 gr. per gal.=2.7 thirty seconds)	
Air bleeds on drains....	(air cocks opened to suit)	
Drain level.....	Float Control	Float Control
Cooler drain outlet temperature	165 degrees	165 degrees
Blow down.....	Constant	Constant
Distiller inlet cooling water from S. W. sanitary system.....	75 pounds	75 pounds
Distiller outlet cooling water pressure.....	(40 pounds to 60 pounds)	60 pounds
Evaporator feed pressure	(20 pounds to 40 pounds)	40 pounds
3-inch unloading valve overboard set at.....	(40 pounds raising—3 pounds at 50 G.P.M.)	
4-inch unloading valve overboard set at.....	(60 pounds raising—4 pounds at 200 G.P.M.)	

Distiller outlet pressure controlled by 4-inch Foster unloading valve overboard. This unloading valve adjustable from 40 pounds to 75 pounds and sufficient pressure will be required to supply the necessary cooling water to sub-coolers and evaporator feed.

Evaporator feed pressure controlled by 3-inch Foster unloading valve overboard. This unloading valve adjustable from 20 pounds to 60 pounds and sufficient pressure will be required to maintain suitable evaporator feed pressure. The setting of this valve to be slightly lower than the setting of the 4-inch valve to insure the necessary cooling water through drain sub-coolers.

Evaporator drain sub-coolers are intended to cool the drains down to about 165 degrees F. when found desirable.

2—Drain Coolers for S.W. Evaporators (Griscom Russell).

Capacity each—4000 pounds per hour—331 degrees to 165 degrees F.

2—Distillers (Griscom Russell).
Capacity each—25 tons (long tons) per 24 hours.
Cooling water—80 degrees F.—ratio of 20 to 1.

2—Aerating Filters (Griscom Russell).
Capacity each—25 tons (6720 gallons) of distilled water per 24 hours.

REFRIGERATING PLANT.

3—Refrigerating Compressors (Carrier Engineering Co.).
CO_2 compressors, vertical—double cylinder—single acting. 5¼ inches diameter × 6-inch stroke 250/327 R.P.M.
Westinghouse motor—115 B.H.P. 250/327 R.P.M.—230 volts D.C.
Refrigerating duty with 10 degrees suction temperature and 85 degrees sea temperature = 24.7 tons per 24 hours.
Refrigerating duty with 10 degrees suction temperature and 88 degrees sea temperature = 23.0 tons per 24 hours.
Air conditioning duty for dining salons with 37½ degrees suction temperature and 85 degrees sea temperature = 71 tons per 24 hours.

2—Cargo Brine Circulating Pumps (Worthington).
Motor driven, horizontal, single stage, centrifugal type.
Capacity—250 G.P.M. against 65 feet total head.
Westinghouse motors—10 H.P. 1150/1750 R.P.M., 230 volts D.C.

1—Miscellaneous Ship's Refrigerator Brine Circulating Pump (Worthington).
Motor driven, horizontal, two stage, centrifugal type.
Capacity—40 G.P.M. against 120 feet total head.
Westinghouse motor—5 H.P. 1375/1750 R.P.M. 230 volts D.C.

2—Condenser Circulating Pumps (Worthington).
Motor driven, horizontal, single stage, centrifugal type.
Capacity—440 G.P.M. against 60 feet discharge head.
Westinghouse motor—12 H.P. 1150/1750 R.P.M. 230 volts D.C.

2—Ice Making Sets (Carrier Engineering Co.).
Capacity each—1,000 pounds per 24 hours.

CARGO BOXES.

	Temperature
4—Cargo spaces and emergency hatch, degrees	10

SHIP'S REFRIGERATED SPACES.

	Temperature
1—Vegetable, degrees	40 to 45
1—Fruit, degrees	40 to 45
1—Meat, degrees	20 to 24
1—Fish, degrees	20 to 24
1—Butter, eggs, milk, degrees	36 to 40
1—Ice cream, degrees	10
1—Poultry, degrees	20 to 24
1—Ice stores, degrees	28

MISCELLANEOUS SERVICE BOXES.

	Temperature
1—3rd class butcher shop, degrees	20 to 24
1—3rd class pantry box, degrees	40 to 45
2—Main galley pantry box, degrees	40 to 45
1—Main galley pastry shop box, degrees	40 to 45
1—Tourist pantry box, degrees	40 to 45
1—Carde manger, degrees	40 to 45
1—Butcher shop box, degrees	20 to 24
1—Service bar box, degrees	40 to 45

Scuttle Butt—cool 390 gallons.

Compressors in Operation for Refrigeration.

Two compressors to have ample capacity to handle the above refrigerating load under usual conditions of cooling down cargo, etc.

One compressor to have ample capacity to maintain above temperatures in boxes after same have been cooled down.

DINING SALON AIR CONDITIONERS.

2—Air Conditioners for Main Dining Salon (Carrier Engineering Company).

2—Air Conditioners for Tourist Dining Salon (Carrier Engineering Company).

Air conditioners have heating coils and are arranged for summer or winter operation and with or without air conditioning.

Summer Operation Without Air Conditioning:

The blowers draw 100 per cent outside air by-passing air conditioning apparatus and discharging air direct to dining salons.

Summer Operation With Air Conditioning:

The air conditioners provide for cooling dining salons with full seating capacity, and the necessary lighting with air conditions as follows:

Outside air—90 degrees dry bulb—78 degrees wet bulb—57 per cent humidity.

Air in dining salons 80 degrees to 82½ degrees dry bulb—50 per cent humidity.

Quantity of air recirculated approximately—75 per cent (automatically controlled).

Refrigeration load for cooling dining salons under extreme weather conditions as outlined above = 68 tons.

One (1) refrigerating machine (standby unit) capable of handling the above load. It is possible to operate more than one refrigerating machine on this duty if desired. The amount of refrigeration necessary is variable depending greatly upon humidity, electric lights, number of people, etc.

Steam pressure to coils—35 pounds.

Pumps and compressors described below.

Winter Operation Without Air Conditioning as follows:

100 per cent outside air capacity of blowers to be heated from 0 degrees F. to 120 degrees F.

2—Blowers for Air Conditioners Main Dining Salon (Sturtevant).

Capacity forward blower 8900 C.F.M. 2.5 inches S.P. 720 R.P.M.

Westinghouse motor forward blower 9.15 H.P. 540/830 R.P.M.—230 volts D.C.

Capacity aft blower 8300 C.F.M. 2 inches S.P., 650 R.P.M.

Westinghouse motor aft blower 6.7 H.P. 490/750 R.P.M.—230 volts D.C.

2—Blowers for Air Conditioners—Tourist Dining Salon (Sturtevant).

Capacity each 4750 C.F.M.—2 inches S.P. 860 R.P.M.

Westinghouse motor 3.8 H.P.—645/985 R.P.M.—230 volts D.C.

2—Pumps for Water Sprays in Main Air Conditioners (Worthington).

Size 2-D-2, Volute pump, horizontal motor driven.

Westinghouse motor—3 H.P. 2800 R.P.M.—230 volts D.C.

2—Pumps for Water Sprays in Tourist Air Conditioners (Worthington).

Size 1½ D. Volute pump, horizontal, motor driven.

Westinghouse motor 2 H.P.—2800 R.P.M.—230 volts D.C

1—Air Compressor (Curtis Pneumatic Machine Company).

Size 2⅛ × 2⅝ and 2⅝ × 2⅝ single cylinder, motor driven compressor.

Maximum discharge pressure,—25 pounds.

Westinghouse motor—¾ H.P.—1150 R.P.M. 230 volts D.C.

Main Steering Gear (American Engineering Company).

Hydroelectric 4 cylinder double ram type.

Angle from hard over to hard over—70 degrees.

Automatic limit stop operated at 1½ degrees past maximum angle of helm.

Time hard over to hard over at 20 knots speed ahead with (1) pumping unit 30 seconds.

Time hard over to hard over at 15 knots astern—74 R.P.M. Propellers—no specified time.

2 pumps—Hele Shaw, variable stroke, rotary type.

Pressure required for maximum ahead torque—585 pounds.

Pressure required for maximum astern torque—1600 pounds.

Test pressure—2200 pounds.

Pressure regulator on pump to prevent overloading of motor.

Storage tank capacity—20 per cent greater than system.

Westinghouse motors—90 H.P. 1000 R.P.M. 230 volts D.C.

Auxiliary Steering Gear (American Engineering Co.).

Motor driven quadrant type.

Angle from hard over to hard over—30 degrees.

Time hard over to hard over at 20 knots speed ahead (20 seconds in P.O.).

Time hard over to hard over at 6 knots speed astern—not specified.

Westinghouse motor—25 H.P. 600 R.P.M. 230 volts D.C.

Windlass (Allan Cunningham).

Horizontal type with twin motor drive.

Hoisting speed for each wildcat with single motor drive will be 5 fathoms per minute when hoisting (1) anchor weighing 21,560 pounds with 30 fathoms of chain.

When heaving both anchors at once under heavy load condition the center clutch on jack shafts to be disengaged, each motor and wildcat operating independently.

One motor can be used on both wildcats only when heaving short under moderate load.

Westinghouse motors—90 H.P. each—400 R.P.M.

30 minutes rating, 230 volts, D.C.

Series wound reversible.

2—Capstans (American Engineering Company).

Motor driven through worm and spur gear with 26 inches diameter gypsy head.

Each capstan to be capable of the following performance:

Pounds Pull	*Rope Speed F.P.M.*
20,000	62
5000	82
Light line load .....(abt. 10 H.P.)	97
Light line load .....(abt. 5 H.P.)	102

Westinghouse motor 75 H.P. 600 R.P.M. 230 volts D.C.

18—Cargo Winches (Lidgerwood).

Motor driven, single drum (18 inches diameter × 20 inches face), single geared, with single winch head. Each winch to be capable of the following performance:

6720 pounds (3 tons)	215 feet per minute
3500 pounds	281 feet per minute
2000 pounds	323 feet per minute
1325 pounds	420 feet per minute
Light line	780 feet per minute

Westinghouse motor 35 H.P. 480 R.P.M. 230 volts D.C.

2—Cargo Winches (Lidgerwood).

Winches and motors same as above except to be arranged for handling 4 tons at a speed of 190 feet per minute.

1—Deck Winch (Lidgerwood).

Motor driven, single drum (12 inches diameter × 16 inches face) double geared, with single winch head.

Capacity 1 ton (2240 pounds) at a speed of 150 feet per minute with single whip.

Westinghouse motor 13 H.P.—775 R.P.M. 230 volts D.C.

THE TREND IN NAVAL ENGINEERING.

ALTERNATING-CURRENT MOTORS AND GENERATORS. THEORY.

BY C. HUEY, CIVIL MEMBER.*

Note by Editor.—This is the ninth of a series of articles on "The Trend in Naval Engineering," the first of which was published in the February, 1931, number of the JOURNAL.

The use of alternating-current has long been common in the commercial world, and, on account of certain inherent characteristics, there is a definite trend toward its application as auxiliary power in Naval ships and in the Marine service in general.

It is therefore desirable from the standpoint of the operating personnel and others concerned with the application in Naval ships, for some one to review the enormous amount of technical literature on the subject of alternating-current generators and motors, their principles of operation and performance, characteristics, the applications of motors to power driven machines, and the other differences from the present well established direct-current practice.

The Direct-Current Navy—It is safe to assert that the U. S. Navy of the present day is a ***direct-current*** Navy. Although the fundamental principles of the performance of direct-current motors are considered to be fairly easy to master, nevertheless there have been many bothersome aspects, such as the causes of sparking brushes, the vagaries of the speed characteristics of the different types of motors under load, the polarity combinations of series, shunt and interpole field coils, the relation of the fields in reversing the direction of rotation of the armature, and the limitations of speed control by variation in field strength.

The Trend to Alternating-Current—The introduction of alternating-current motors for auxiliary power applications on certain types of Naval ships has now presented new problems to the

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direct-current-minded personnel. The self appointed task of the writer is to clear away some of the mysteries of alternating current, by simplified explanations of the fundamentals of alternating current and the characteristics of motors of the types likely to be installed for various purposes.

*Direct-Current Fundamentals**—The explanations of *alternating-current* power will be based as much as practicable on comparisons with *direct-current*, or with mechanical power. The relations of the variables of any *direct-current* circuit based on Ohm's and Joule's laws ($E = IR$; $W = EI$; $W = I^2R$) are illustrated (not to scale) in Figure 1, in which E = electromotive force in volts, I = current in amperes, R = resistance in ohms and W = power in watts.

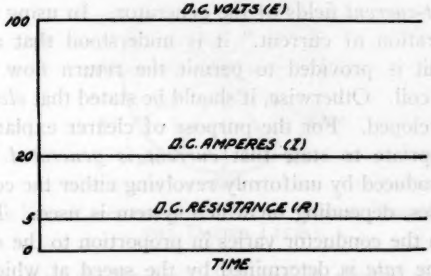

FIG. 1.—THE RELATION OF DIRECT-CURRENT VOLTS, AMPERES AND RESISTANCE.

In the example given

$W = EI = 100 \times 20 = 2000$ watts or $W = I^2R = 400 \times 5 = 2000$ watts

2000 watts $= \frac{2000}{1000}$ kilowatts or $\frac{2000}{746}$ horsepower, (*e. g.* motor input).

The current in the D.C. circuit illustrated is unidirectional, continuous, and unvarying at every instant of time. Volts and amperes may be measured with direct-current meters, directly in

* Proofs of the theories and formulae used in this article may be found in "Alternating Current and Alternating Machinery" by D. C. & J. P. Jackson, "Applied Electricity for Practical Men" by Arthur J. Rowland, and other text books.

terms of their values. The ohmic or total resistance of the circuit invariably is the terminal volts divided by the amperes flowing in the circuit. The factors, inductance or electrostatic capacity, play no part in direct current circuits such as they do in alternating current circuits, except momentarily at the start and at the interruption of the circuit, if inductance or capacity, or both, are present.

Alternating-Current Fundamentals—The explanations and illustrations of alternating current circuits may best be presented by starting with the basic characteristic, the relative value of the current (or voltage) at different instants of time. In the *synchronous* generation of alternating current of the form with which we are chiefly concerned, the current is produced by a coiled conductor alternately cutting the magnetic flux of the north and south poles of the *direct-current* fields of the generator. In using the expression "generation of current," it is understood that an external closed circuit is provided to permit the return flow of current through the coil. Otherwise, it should be stated that *electro-motive force* is developed. For the purpose of clearer explanation, it is more appropriate to state that *current is generated.* The flux cutting is produced by uniformly revolving either the conductor or the field poles, depending on which system is used. The current generated in the conductor varies in proportion to the *rate* of flux cutting. The *rate* is determined by the speed at which the conductor (or the field) rotates, and the *amount* of flux. As an alternator-generator has no rectifying commutator as in the case of a direct-current generator, the current generated rises, falls, reverses and rises in opposite direction, as the coil passes through the magnetic flux of the alternate north and south poles of the direct-current fields. This generation of A.C. is known as *synchronous* generation as it is directly dependent on, and is in synchronism with the mechanical rotation of the conductor coils, or the fields. Direct-current is generated in exactly the same way, except that by the use of a commutator to which the *individual* coils are connected, the current in the *external* circuit always flows in the same direction.

The Wave Form of Alternating-Current—The wave-form or wave shape of the current generated in a well designed alternator

is sinusoidal, or shaped similar to a *sine wave*. Commercial standards require that the wave shape of an alternator shall not depart from the true sine wave by more than 10 per cent. The sine, which is commonly used in alternating-current work, is graphically shown in Figure 2, the *sines* of the various basic polar angles being ½ of the dotted lines. The term "angles" used in the text refers to the angular distance from one reference point in the field polarity to the next corresponding point. For example, from the center of a north pole to the center of the next north pole is referred to as 360 degrees. The cycle of effects repeat in terms of 360 degrees for every pair of north and south poles.

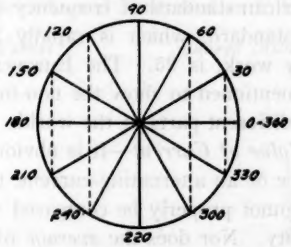

FIG. 2.—THE SINE VALUES OF BASIC POLAR ANGLES (SHOWN AS ONE-HALF THE DOTTED LINES).

The values of the "sines" of Figure 2, plotted as in Figure 3, furnish the outline of the alternating-current sine wave that is generated at the electrical angles between pairs of poles of the alternator. The current from 0-180 degrees being in one direction (positive) and from 180 to 360 degrees, in the opposite direction (negative).

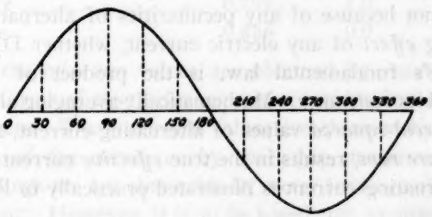

FIG. 3.—THE SINE VALUES PLOTTED FOR ONE COMPLETE SINE WAVE.

The Frequency of Alternating-Current—The *frequency* with which the complete wave as shown occurs in one revolution of an alternating-current generator is directly dependent on the number of *pairs* of poles (one north and one south) of the generator and the speed of the rotating member. The frequency, cycles per second, is obtained by multiplying the number of pairs of poles by the revolutions *per second* of the revolving field or revolving armature. For example, in a generator revolving at 3600 revolutions per minute (60 per second), one north and one south pole would produce 60 cycles A.C. per second. Likewise, if revolving at 1800 R.P.M. 2 north and 2 south poles would also produce 60 cycles per second; similarly for 900 R.P.M. and 4 pairs of poles. The accepted American standard of frequency (cycles per second) is 60. The old standard, which is rapidly becoming obsolete, except for railway work, is 25. The European standard is 50. These points are mentioned to show the non-interchangeability of A.C. apparatus in different parts of the world.

The Effective Value of Current—It is obvious from a study of the sine wave shape of an alternating-current, that the "peak" or maximum value cannot properly be compared with direct-current of the same intensity. Nor does the *average* of the instantaneous values compare with equivalent direct-current, unless the alternating loops have a theoretical rectangular shape, in which case the maximum, average, and effective values, would be equal to each other. The effective or useful current in alternating-current circuit is the *square root of the average of the squared instantaneous values*. This *effective* or useful current gives the same heating effect as direct-current of the same intensity flowing in a circuit having the same resistance. The *effective* value is always less than the *maximum* value and is appreciably higher than the *average* value, but not because of any peculiarities of alternating-current. The heating *effect* of any electric current, whether D.C. or A.C., from Joule's fundamental law, is the product of the *current squared* and its resistance. Mathematically averaging all the instantaneous *current squared* values of alternating-current, and extracting the *square root,* results in the true *effective* current. This feature of alternating-current is illustrated practically in Figure 4.

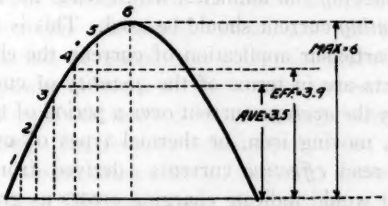

FIG. 4.—PRACTICAL RELATION OF THE AVERAGE, EFFECTIVE, AND MAXIMUM VALUE OF HALF A SINE WAVE.

By the use of arbitrary (but incorrect for a sine wave) values for current as in Figure 4, the *mathematical* comparison of the *average* and *effective* values is illustrated as follows:

Instantaneous Values	*Squared Instantaneous Values*
1	1
2	4
3	9
4	16
5	25
6	36
$6 \mid 21$	$6 \mid 91$
$I = 3.5 =$ *average*	$I^2 = 15.2$
	$I = 3.9 =$ effective

For true sine waves, the mathematical relation of the *average, effective* and *maximum* values of volts or current shown in the following tabulation from which if any one is known, the other two may be evaluated.

Maximum	1.0	1.41	1.57
Effective	0.707	1.0	1.11
Average	0.636	0.9	1.0

In practice the *effective* values are obtained by *readings of* the commercial forms of electrodynamometer, moving iron, or thermal types of meters. The *maximum* values are obtained by evaluating oscillograms of the curves or by spark-gap measurement. The *average* readings are not obtainable by meter readings of true alternating-current. However, it is to be noted that in storage battery charging from *rectified* alternating-current, a direct-current *perma-*

nent magnet moving coil ammeter, which reads the *average* of the resultant *pulsating* current should be used. This is for the reason that, in this particular application of current, the electro-chemical charging effects are in terms of the *quantity* of current which is represented by the *average* current over a period of time. Electro-dynamometer, moving iron, or thermal types of meters for this work, which read *effective* currents (derived from the *current squared* laws) would indicate charging errors as great as plus 57 per cent, depending on the wave form of rectification.

Ammeters Indicate Effective Current—All of the conventional (Navy) types of alternating-current ammeters read in terms of *effective* current. The readings so obtained may be correctly used for obtaining the values of the effective current in any part of 60 cycle alternating-current machines or circuits.

The Effective Voltage—The statements made above with respect to current hold also for the alternating voltage (electromotive force) in regard to the development of the *sine* wave, the *effective* values, and the measurements by means of the conventional (Navy) types of alternating-current voltmeters.

The foregoing briefly outlines the fundamental similarities and differences of direct-current and alternating-current. The momentary lag at the closing of a *direct current* circuit containing inductance (magnetic fields which retard building up of the current) likewise holds for A.C. circuits. In the rapid reversal of current in A.C. circuits, the retarding effects are more or less continuous, in amount depending on the *frequency* of the alternations. The *principles* which relate to the flow of electric currents, whether direct or alternating, and which are applied to the design and construction of machines and circuits, are the same.

Phase Relation to Power—The characteristic difference between direct-current and alternating-current, in addition to the alternating wave form of the latter, is in the phase or time relation between the current and the voltage which produces the current. In *settled* direct-current circuits the voltage and current are always in phase and have the same *time* period under all settled conditions of the circuit. The power under settled conditions is $W = EI$. In alternating-current circuits, the *instantaneous* power is likewise $w = ei$, e and i being instantaneous values of voltage

and current. The useful power in practical applications is the *average* power of the alternating voltage and current waves, as obtained by the use of wattmeters which integrate the cycles of instantaneous power in terms of continuous power. The product of the effective volts and amperes is the average power, and hence the useful power. The term *effective power* is not used in alternating-current nomenclature; it is, however, synonymous with the *average power*. The expression W = EI holds true for alternating-current only when the voltage and current are *in phase* as in the case of a circuit made up of ohmic resistance only (containing no inductance or electrostatic capacity). When *in phase*, the current and the voltage waves pass through zero and reach their maximum values at the same instant.

Current and Volts When in Phase—The relation of alternating voltage, current and power, when *in phase*, is shown for one complete wave, in Figure 5.

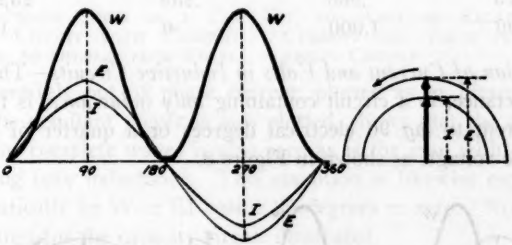

FIG. 5.—POWER CURVE OF A RESISTANCE CIRCUIT WITH CURRENT IN PHASE WITH VOLTS.

FIG. 6.—ANGULAR RELATION OF CURRENT AND VOLTS IN A RESISTANCE CIRCUIT.

The "in phase" angular relation of the current and voltage is shown in Figure 6. It will be noted in Figure 5 that the power curve, which is the product of E and I, is plotted entirely on the positive side of the base line. This is for the reason that the power so derived is indicated as "useful" regardless of the direction of the "in phase" voltage or current, also since the product of two positive or two negative factors is positive. The *power-factor* in this illustration is *unity*, and may be indicated by the formula $\frac{W}{E\,I} = 1$. The unity power-factor is sometimes expressed in per cent, *i.e.*, as 100 per cent power-factor.

Power-Factor—The term *power factor* in alternating-current fixes the value of the electrical angle or phase displacement between the voltage and the current. The value of the angle is obtained by reference to trigonometric tables of *functions* of angles. The *function* which applies in this problem is known as the *cosine* of the angle. The cosine of any angle A, equals the sine of (90 degrees — A). The sines, cosines, and the corresponding power-factors in per cent of a number of angles are tabulated as follows:

Angle	*Sine*	*Cosine*	*Per Cent Power Factor*
0	0	1.000	100.0
15	.259	.966	96.6
30	.500	.866	86.6
45	.707	.707	70.7
60	.866	.500	50.0
75	.966	.259	25.9
90	1.000	0	0

Relation of Current and Volts in Inductive Circuits—The effect of inductance, in a circuit containing *only inductance*, is to cause the current to *lag* 90 electrical degrees or a quarter of a cycle after the voltage, as shown in Figure 8.

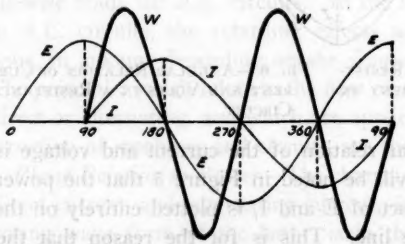

FIG. 7.—POWER CURVE OF AN INDUCTIVE CIRCUIT WITH CURRENT LAGGING 90 DEGREES FROM VOLTS.

FIG. 8.—ANGULAR RELATION OF CURRENT AND VOLTS IN AN INDUCTIVE CIRCUIT.

With entirely out of phase current, plotted as in Figure 7, in which the resultant power is also plotted, the average power for complete waves is zero. The out of phase relation of voltage and

current is represented also in Figure 8. The power for this condition is expressed by the equation W = EI cosine 90 degrees. The cosine of 90 degrees being zero, the value of the equation becomes zero and no power is indicated for the inductive circuit illustrated.

Relation of Current and Voltage in Capacity Circuits—The effect of capacity, in a circuit containing *only capacity*, is to cause the current to *lead* 90 electrical degrees, or a quarter of a cycle, ahead of the voltage, as shown in Figure 10.

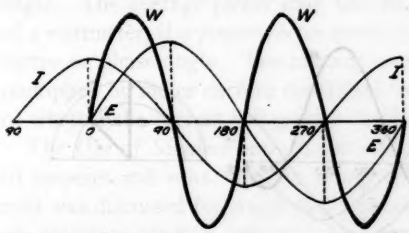

FIG. 9.—POWER CURVE OF A CAPACITY CIRCUIT WITH CURRENT LEADING 90 DEGREES FROM VOLTS.

FIG. 10.—ANGULAR RELATION OF CURRENT AND VOLTS IN A CAPACITY CIRCUIT.

This entirely out of phase current, plotted as in Figure 9, in which the resultant power is also plotted, shows that the average power for complete waves is also zero as is the case with circuits containing only inductance. This condition is likewise expressed mathematically by W = EI cosine 90 degrees = zero. No power is indicated for the capacity circuit illustrated.

Explanation of Zero Power—In the foregoing illustrations of circuits in which the current is entirely out of phase with the voltage, as when the circuit contains only inductance, or only capacity, the circuit *absorbs* power when the ordinates are positive, and when negative, the circuit *gives out* energy which was stored as magnetic field in the *inductive* circuit, or, as a condenser charge in the *capacity* circuit. The net result is zero, comparable with the work done by a swinging pendulum.

Importance of Power-Factor—The foregoing illustrations are given to show the extreme limits of lagging or leading current and establish the importance of the *power-factor* in measuring A.C. power. In actual practice when dealing with motors of the asynchronous types, such as *induction motors, repulsion motors,* and

series motors, which revolve more or less independently of the frequency of the impressed voltage, the circuits are a combination of resistance and inductance.

Effect of Resistance and Inductance—The effect of the combination of resistance and inductance is to produce a lagging current of some value less than 90 degrees, depending on the relative values of resistance and inductance. In Figure 11, the voltage current, and power are plotted to illustrate a lagging current of 45 degrees as indicated in Figure 12.

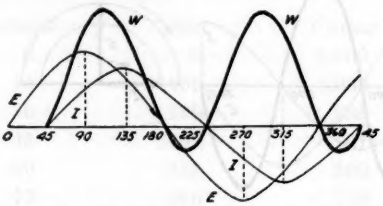

FIG. 11.—POWER CURVE OF A RESISTANCE AND INDUCTIVE CIRCUIT WITH CURRENT LAGGING 45 DEGREES FROM VOLTS.

FIG. 12.—ANGULAR RELATION OF CURRENT AND VOLTS IN A RESISTANCE AND INDUCTIVE CIRCUIT.

In the above illustration, the current, being only partly out of phase—45 degrees, some negative power is produced as shown in the plotted power curve in Figure 11. Mathematically $W = EI$ cosine 45 degrees $= EI\ 0.70$. In any circuit if the values of W, E and I are known, the power-factor may be determined by the equation, cosine phase angle = Power Factor $= \frac{W}{E\ I}$. In power plant distribution it is the practice to add a suitable amount of *capacity* to the system, usually in the form of over excited synchronous motors, so that the effect of inductance is reduced. By adjusting the plant, if practicable, to obtain unity power-factor, the least current flows for the power output.

Reading of Volts, Amperes, Watts and Power Factor—In all of the foregoing illustrations of circuits containing (a) only resistance (such as incandescent lamps, heaters and plain rheostats), (b) only inductance (a theoretical circuit), (c) only capacity (such as an electrostatic condenser), (d) resistance and inductance (such

as asynchronous motors, or magnetic coils of appreciable ampere-turns), the effective voltage or current, regardless of wave shape, is correctly indicated on all commercial types of electrodynamometer, moving iron, or thermal instruments. The average power of *resistance* circuits can be obtained from the product of the meter readings of voltage and amperes. For other circuits the average power in watts may be obtained from any commercial type of wattmeter which takes into consideration the power factor or phase angle. The average power may also be obtained without the use of a wattmeter, if a *power-factor meter* is used to obtain the power-factor or phase angle. The average power then is, meter voltage multiplied by meter current multiplied by power-factor, regardless of whether the current is lagging or leading.

The Use of Single-Phase Power—In the foregoing the relation of amperes and volts, with the resulting power, in alternating-current was discussed for *single-phase* circuits as constituted by a single armature winding delivering its generated power to an external circuit through the medium of *two* wires or conductors. Single-phase power, because of the minimum of wires required for service, is still widely used for power driven devices of the portable, domestic, and hand tool types such as vacuum cleaners, blowers, mixers, fans, drills, and grinders.

In addition to the simplicity of the single-phase power circuit, a fixed direction of rotation of motors of the above classes is necessary, and can be assured on single phase motors regardless of the polarity the operator establishes in the circuit. With *two* and *three* phase motors this is not the case. Also, due to the economic necessity of manufacturing the small and fractional horsepower motors to fit either A.C. or D.C. power, a type of single phase, 60 cycle A.C. motor has been developed known as the ***universal current*** motor, which operates equally well on direct current of the same voltage.

Single-Phase Circuits—The generation of single phase, 60 cycle power, is portrayed in a schematic diagram in Figure 13, the simplicity of which is shown in the single sine wave and the two connections of the generator coil to the two terminals of the motor winding.

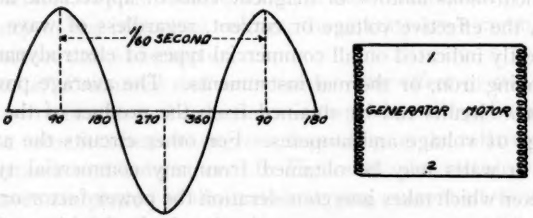

FIG. 13.—SINGLE SINE WAVE 60 CYCLE CURRENT, AND DIAGRAMMATIC GENERATOR AND MOTOR CONNECTIONS.

Two-Phase Power—As was to be expected, the simplicity of single phase power held sway only long enough for inventive economists to discover inactive and vacant spaces in the armature core of the single phase generators. To this they added a duplicate set of windings, displaced 90 electrical degrees from the first. The angular relation of the two duplicate windings, generating *two-phase* current, known also as the *quarter-phase* current, is shown in Figure 14.

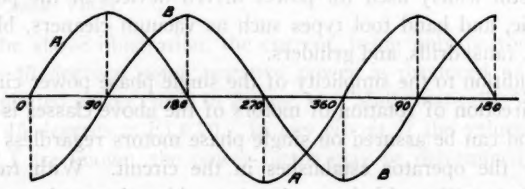

FIG. 14.—THE 90-DEGREE PHASE RELATION OF TWO-PHASE, OR QUARTER-PHASE, SINE WAVE CURRENTS.

The addition of the second winding to the generator brought with it additional schemes for distribution and use of the power for circuits and motors. For the same reasons as in the case of the generator, the two-phase winding is used on two-phase motors with resultant increase in power output for the same size and weight. By the use of the two-phase system, the flexibility of distribution is increased by the fact that not only can two-phase motors be employed, but either of the phases can be used separately for single-phase work. Figure 15 shows in diagrammatic

form, a two-phase generator and the connections to two single, and one two-phase motors, using four wires.

Measurement of Two-Phase Power—The measurement of the total power in a two-phase, *four* wire circuit, regardless of wave form or power factor, is obtained from the arithmetical sum of the wattmeter readings in each circuit. When there is an interconnection between the two phases, existing in special instances, this rule does not apply. In the case of a two-phase motor, it may be assumed that the phases are in balance. The power then is read

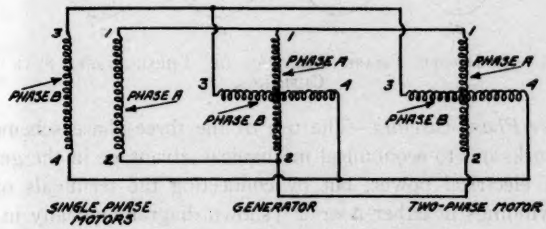

FIG. 15.—DIAGRAMMATIC CONNECTIONS OF A TWO-PHASE, FOUR-WIRE GENERATOR, TO SINGLE AND TWO-PHASE MOTORS.

as twice the wattmeter reading of either phase. Two-phase generators may be connected in such manner that one of the four basic terminals is in common in the two phases. This results in a *three* wire system of distribution, the power of which is read somewhat differently from the above rule. Due to the fact that two-phase power has practically been superseded by *three*-phase power, *two*-phase power is discussed here only to show the evolution to *three*-phase power.

Three-Phase Power—The conception of the three-phase generator is the result of the practical employment of the inactive parts of the single-phase generator and the remaining inactive parts of the two-phase generator. By the use of a triplicate set of windings on the armature core, spaced 120 electrical degrees from each other, three-phase current is generated in the same manner as in the case of three separate single phases, except that the instantaneous values of the current in the three circuits are 120 degrees apart as shown in Figure 16.

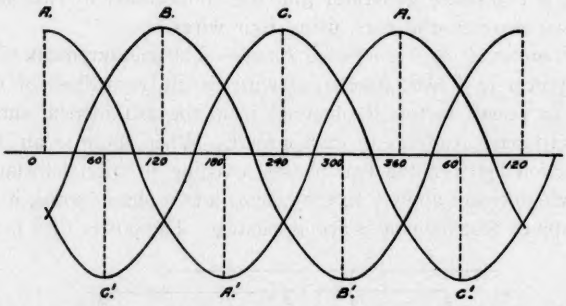

FIG. 16.—120-DEGREE PHASE RELATION OF THREE-PHASE, SINE WAVE CURRENTS.

Three-Phase Circuits—The use of the three-phase scheme not only works out to economical mechanical advantage in the generation of electrical power, but by connecting the terminals of the three windings in either Δ or Y (shown diagrammatically in Figures 17a and 17b) the number of electrical leads in the three-phase system is reduced from six to three. In both of these circuits, the current *leaving* any one terminal, at any instant, is equal to the sum of the currents *entering* the other two terminals. This is in accordance with Kirchoff's law of current flow which likens current to an incompressible fluid; this means that the current *leaving* a coil must at all times be *exactly* equal to the current *entering* the coil.

FIG. 17a.—DIAGRAMMATIC DELTA CONNECTION.

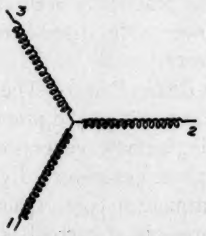

FIG. 17b.—DIAGRAMMATIC Y CONNECTION.

Delta Three-Phase Circuits—In Figure 17a, the arrangement is known as the *Delta* connection. The *voltage* between any of two of the 3 terminals, as may be seen by inspection of the diagram,

is the voltage generated by one coil. The *current,* however, in any of the 3 line wires, is a combination of those in two adjacent coils. In terms of current in one coil, the line current in any leg in a balanced system is $\sqrt{3}$ I or 1.73I, where I is the current in a coil.

Y Three-Phase Circuits—In Figure 17b, the arrangement is known as the Y connection. The *voltage* between any two of the three terminals, as may be seen by inspection of the diagram, is the vector sum of the voltages generated by two coils displaced by 120 electrical degrees.

In terms of the voltage of one coil in a balanced system, it equals $\sqrt{3}$ E or 1.73E where E is the voltage of one coil. In the case of the current, the line current will be the *same* as the current through which the line terminates.

The Uses of Three-Phase Delta and Y Circuits—It is obvious from the above that in the *Delta* connected generator the *current* available in any line may be 1.73 greater than rated for any one coil, the voltage being the generated voltage of one coil. In the *Y* connection, the *voltage* available is 1.73 greater than generated in any one coil, the current available being the normal current rating of one coil. These features of 3 wire three-phase generators, applying also to motors and power from transformers, gives a certain amount of flexibility in the choice of two current or voltage ratings, for starting and application purposes. In either the Delta or the *Y* connected generators, single-phase current is available from any pair of line wires for use as single-phase power.

Measurement of Three-Phase Power—The measurement of total power in a 3 wire, three-phase motor or other device, regardless of wave form, or power factor, or whether it is Delta or Y connected, can be made by so many methods, that unless the explanations are limited to methods for which suitable instruments are likely to be available, the problems presented are probably the most confusing of all the alternating-current power measurements. For example, there are methods using three single-phase wattmeters, two single-phase wattmeters or one single-phase wattmeter, methods requiring an artificial neutral or a Y box, and methods requiring access to the three-phase net work *before* they join the line wires. The following methods have been selected as being the most practicable and apply to both the Delta or Y circuits.

One Wattmeter for Measurement of Three-Phase Power—In a *balanced* circuit where the power-factor is equal in all branches and the load is uniformly distributed, as assumed in a three-phase motor, and where the load does not vary, one wattmeter may be used by connecting the current coil in one line, and connecting the free terminal of the voltage coil alternately to the other two lines. See Figure 18. The sum of the two readings gives the total power of the three phases.

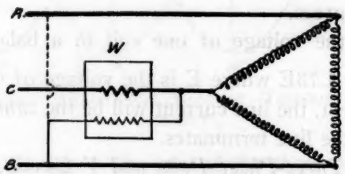

FIG. 18.—MEASUREMENT OF POWER IN A THREE-PHASE BALANCED CIRCUIT, USING ONE WATTMETER.

Two Wattmeters for Measurement of Three-Phase Power—With the use of two single-phase wattmeters as connected in Figure 19, the power in a three-phase circuit may be measured with entire independence of the balance of the phases, or of power-factor. When the power-factor is *greater* than .50 the two wattmeters should read in the positive direction and the sum of the readings will give the total power. However, when the power-factor in the circuit is *less* than .50, as may be the case in partially loaded induction motors, one of the wattmeters should read negative or reversed, due to the relation of the currents in its current and voltage coils. The value of the negative reading is obtained by reversing the voltage connection in the wattmeter so that a *positive* reading is obtained. This reading is then subtracted from the reading of the other wattmeter and the result is the total power.

The Double or Three-Phase Wattmeter—In the use of the two wattmeter method described in the previous paragraph, the difficulty in determining the conditions of the circuit which will or will not give negative readings on one of the wattmeters, make the use of the *double* wattmeter, known as the three-phase wattmeter, very desirable as a portable instrument and indispensable as a switchboard instrument. This instrument consists of two movable coils on one spindle, and two fixed field coils in relation to the movable

coils, equivalent to two single-phase wattmeters, except that they are in mechanical combination. Such an instrument sums up the double reading *algebraically,* and a single reading gives the total power direct.

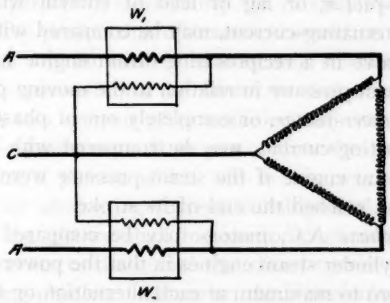

FIG. 19.—MEASUREMENT OF POWER IN A THREE-PHASE UNBALANCED CIRCUIT, USING TWO WATTMETERS.

Comparisons of Electric Power with Mechanical Power—It sometimes helps in the conception of electric power, to compare it with well known forms of mechanical power. The analogies, of course, are not strictly correct, but the pictures presented show many similarities. For instance—

(a) *Electric current* in a circuit has features of *incompressibility* such as possessed by liquids.

(b) *Direct-current* may be compared with the flow of water from an elevation, the electro-motive force or voltage being analogous to "head."

(c) *Ohmic resistance* in wires which may be compared with water *friction* in pipes, is a function of diameter and length of wire as the friction is a function of diameter and length of pipe.

(d) *Time lag* in the start of direct-current in inductive circuits, followed by the steady flow, and the tendency to continue after the circuit is interrupted, may be compared with *inertia* and *momentum* when a water valve is open or closed.

(e) *Power* in a direct-current circuit in terms of volts and amperes may be compared with the power of flowing water; thus an increase in volts or amperes, or an increase in hydrostatic head or quantity of water, increases the power.

(f) *Alternating-current* may be compared with the steam flow controlled by the slide-valve of a double acting reciprocating engine, the current like the steam being useful in both directions of motion.

(g) *Power-factor,* or lag or lead of current with respect to the volts in alternating-current, may be compared with lag or lead of the slide valve in a reciprocating steam engine in the time of delivery of steam pressure in relation to the moving piston.

(h) *Zero power-factor,* or completely out of phase current and volts in alternating-current, may be compared with the effect on power in a steam engine if the steam pressure were applied only after the piston reached the end of its stroke.

(i) *Single-phase* A.C. motors may be compared with double acting *single* cylinder steam engines in that the power in both cases varies from zero to maximum at each alternation or stroke. Also, similar to the single cylinder engine, certain types of single-phase A.C. motors, have " dead centers" and will run in either direction if forced over the " dead center."

(j) Two-phase or quarter-phase A.C. motors may be compared with double acting, *two* cylinder, steam engines, with cranks at 90 degrees. The flow of power or torque of the two-phase motor, however, is fundamentally *continuous,* like the *direct-current* motor, or a steam turbine.

(k) The three-phase A.C. motor may be compared with the double-acting *three* cylinder steam engine with cranks at 120 degrees. The flow of power or torque of the three-phase motor, however, is fundamentally continuous like the direct-current and the two-phase motor.

Alternating-Current Versus Direct-Current Applications—In the substitution of alternating-current for direct-current in electrical power installations, what are the advantages of the substitution? The answer to this question is by no means simple nor can it be conclusive from every standpoint. The scope of the answer may be narrowed by eliminating from consideration, railway transportation and long distance transmission. For Naval and marine service in general, one of the outstanding advantages in the use of alternating-current is not obtained. This is in the enormous saving in copper in the generation, switching, transformation and trans-

mission by the use of very high voltages with the corresponding lowering of the current transmitted. What then are the advantages of using alternating-current in closely connected power installations? It cannot be conceded that *direct-current* has failed entirely or in part, to be satisfactory for Naval purposes. Neither can it be readily proven that *alternating-current* on shipboard will be a "cure-all" or an entirely satisfactory substitute for direct-current. The answer must be found in evaluating the relative merits of the two systems after outlining their respective advantages and disadvantages. It is not within the province of the writer to paint a picture which will in any way discourage the introduction of alternating-current on shipboard. Neither is it desired that the picture shall indicate predominating features of a'ternating-current to the extent that direct-current should be discarded as being obsolete. However, the situation as it appears at the present writing indicates the adoption of a middle course where alternating-current will perhaps be given a try-out for its equal or superior adaptions while direct-current will be coordinated with it and will be retained for purposes where there is no equivalent substitute.

The Advantages of Direct-Current—

(a) *Generation.*

1. Generators are self exciting, *i.e.,* require no auxiliary excitation.

2. Generators can be made self compounded to any desired degree so that normal volts are delivered at the power terminals regardless of capacity load demand.

3. Changes in the basic speed of the prime mover may be compensated for by rheostatic voltage control, and the generator caused to deliver normal electric power.

4. Generators may be paralleled at different basic speeds and regardless of the instantaneous angular position of the two armatures.

5. The direct-current generated is necessary for direct storage battery charging, the operation of carbon arc searchlights, and the exciting fields of synchronous alternating-current generators.

6. The power-factor is always unity, *i.e.,* 100 per cent useful current.

(b) *Transmission.*

1. No dielectric losses. This advantage is negligible for low voltages.

2. Only 70 per cent of the line insulation required for the same effective volts as alternating-current, the latter being insulated for the maximum or peak voltages. Cable designers usually ignore this for low voltages.

3. Two wires only to be insulated compared to three wires of three phase current.

4. No inductive or electrostatic capacity troubles. The capacity advantage is negligible for low voltages.

5. Single cables, or duplex cables, armored or unarmored, may be installed without regard to heating of bulkheads from magnetic induction. For A.C., special arrangements are necessary.

(c) *Motors and Control.*

1. The starting current of all types of D.C. motors is less than for A.C. squirrel-cage and synchronous motors. The starting, running and overload torques of the series and compound motors are of high order.

2. The speed range of shunt motors by field control which gives practically constant speeds at higher than normal values with negligible rheostatic losses, has no equivalent in A.C. motors, except the type known as the brush-shifting motor.

3. The operation of magnetic brakes and dynamic braking is satisfactory for D.C. motors. Special schemes are necessary for the equivalent A.C. magnetic brake. A.C. dynamic braking is available only on synchronous motors.

4. The no load current of D.C. motors is only about 5 to 10 per cent, compared with A.C. motors of 20 to 40 per cent.

5. The various protective devices of control panels such as the undervoltage, no voltage, overload coils, magnetic blowout coils, do not require laminations or shading coils as in A.C. There is less heating in D.C. devices, which is favorable for water-tight enclosures.

6. A large number of speed control points is more practicable for D.C. than for A.C.

7. Where speed control is required, space requirements are smaller than for A.C.

The Disadvantages of Direct-Current—

(a) *Generation.*

1. The main current being generated on the revolving unit, involves many detailed construction features in regard to the coils and commutator. This implies appreciable cost, weight, and upkeep.

2. The main generated current requires rectification by means of an elaborately insulated commutator with the addition of commutating brushes, brush holders, springs, insulation, rocker arm, etc.

3. Special commutating poles, with main current windings, are required to obtain satisfactory commutation at all loads.

4. Mechanical brush friction and electrical losses in the brushes adds to the heat and plant losses.

5. Upkeep of the commutator and brushes may be of appreciable amount.

(b) *Transmission.*

1. D.C. power, in general, is usually transmitted with two separate cables. This requires additional water-tight stuffing tubes for bulkheads, etc., compared to the single triple-conductor cable usual for A.C. power.

2. The direct-current magnetic fields surrounding large cables require special shielding of electrical indicating instruments.

(c) *Motors and Controls.*

1. All forms of D.C. motors require main current commutators and brushes which increase the weight, cost, up-keep, and require accessibility in the design of enclosing covers.

2. Special commutating poles with main current windings are required to give satisfactory commutation at all loads.

3. There is no direct-current substitute for the synchronous A.C. motor which delivers constant synchronous speed at all loads and regardless of its temperature.

4. All types of D.C. motors over one horsepower require starting resistors to reduce the starting current, to reduce the starting torque, and to reduce the burning of the commutator and brushes.

The Advanages of Alternating-Current—

(a) *Generation.*

1. The *main current* windings of synchronous alternators may be, and usually are, the stationary unit. Electrically and mechanically this is a more economical construction than in the case of direct-current where the main current windings revolve.

2. The *field* windings of synchronous alternators may be, and usually are, the revolving unit. No commutator is required in this construction as the relatively small field current is applied by means of a few brushes and two collector or slip rings.

3. As no commutator, no series fields, or no interpole fields are required, a minimum of connections, terminals, and insulated parts are to be maintained.

(b) *Transmission.*

1. The magnetic fields produced by flow of current are inherently negligible in their effect on electrical instruments, by virtue of the alternations.

2. Various voltages, either higher or lower than the generated voltage, may be transmitted by the use of transformers.

3. Triple-conductor cables usually used for A.C. power are, in general, more economical of space than the two-conductor cables used for D.C. power.

(c) *Motors and Control.*

1. The *main current* windings of the principal types of motors such as the squirrel-cage induction motor and the synchronous motor are stationary, and do not require commutators.

2. The slightly drooping speed-load characteristics of induction motors, unaffected by temperature rises, is favorable for use with impeller pumps and centrifugal fans.

3. The constant speed characteristic of synchronous motors which operate in synchronism with the line frequency, when required for special applications, have no substitute in D.C. motors.

4. Induction motors having no main current commutators, may be started directly across line voltage. This feature is limited only by the capacity of the plant to properly handle the current demand.

5. By the use of auto-transformers, various starting and running voltages may be obtained for certain types of motors, with negligible external losses.

6. Squirrel-cage motors are inherently rugged. They have only two wearing parts, *i.e.*, shaft and bearings.

The Disadvantages of Alternating-Current—

(a) *Generation.*

1. The *inherent* voltage regulation of synchronous generators is very poor. It is not practicable to use the main current to serve for additional excitation similar to the series fields of D.C. generators.

2. To obtain normal generated voltage at various loads, auxiliary automatic regulators are required which operate by varying the field excitation.

3. Synchronous generators are not self exciting as is the case with D.C. generators. They must be separately excited by direct-current, generated by auxiliary D.C. generators.

4. In order to deliver constant normal frequency, the steam governor of the prime mover is required to operate to a higher degree of perfection than with direct-current.

5. Synchronous generators in parallel share the load in relation to the operation of the steam governor of the prime movers. There is no equivalent to the equalizer connections or shunt-field rheostat operating on direct-current generators which serve to maintain and regulate the load balance. A.C. generators in parallel, however, hold together electrically, by interchange of synchronizing currents.

6. The power-factor usually obtained in operating plants is usually not over 80 per cent. This results in additional copper losses over direct-current.

7. The paralleling of two or more synchronous generators, especially on small installations with relatively large motors, is more difficult than with direct-current. In every case, the speed of the prime movers requires manipulation to obtain mechanical and electrical synchronism in order to obtain the initial electrical union.

8. Unbalanced phases, which usually obtain in installations where there is considerable number of single-phase motors, battery rectifiers, unbalanced lighting demands, etc., result in considerable variation of line volts between phases which cannot be compensated by rheostatic control of the generator exciting field. The use of external *phase balancers* for this purpose adds additional devices not required by direct-current distribution.

(b) *Transmission.*

1. The insulated cables theoretically require higher insulation values than the same effective voltage of direct-current in the proportion of 1.41 to 1, for the reason that the insulation is required to withstand the peak of the A.C. voltage waves.

2. Dielectric losses are always in evidence with A.C. distribution in forms which do not exist in direct-current. Certain forms of dielectric losses lead to break-downs in the insulation.

3. Hysteresis and eddy current losses, which are not in evidence with direct-current, may be of serious amount if the direct-current lay-out is used for A.C., or if certain special precautions are not used in wiring A.C. power.

4. High induced voltages, which may be destructive, are possibilities accompanying alternating-current installations. These voltages may interfere with communication circuits.

(c) *Motors and Control.*

1. Synchronous motors, which have ideal constant speed characteristics, are handicapped by requiring direct-current excitation.

2. Most types of single-phase motors and some types of variable speed polyphase motors require commutators in which the sparking troubles are greater than with direct-current commutators.

3. There is no A.C. motor, except the brush-shifting type, that has speed-load characteristics equivalent to the shunt field controlled D.C. motor.

4. There is no A.C. motor which gives the exact equivalent in heavy starting torque of the D.C. series or heavily compounded D.C. motor.

5. A.C. motors have definite overload limits beyond which they stop running. D.C. motors have practically unlimited torque.

6. Variable speed, wound-rotor, induction motors, require collector rings, brushes and upkeep. The rheostatic losses in this form of speed control is of considerable amount comparable with main current resistors in D.C. speed control.

7. The most important class of motors is the induction type. This type is designed with very small air-gap between rotor and stator in order to obtain satisfactory electrical efficiency. Bearings require more frequent renewal in order to maintain running clearance.

8. Magnetic brakes, in general, are not effective with alternating-current.

9. There is no equivalent in A.C. motors of the electro-dynamic braking produced by short circuiting the armature with full excitation, except the synchronous motor.

10. All iron cores used in motors and control are required to be laminated to prevent magnetic heating, and rigidly assembled to prevent mechanical noises.

11. High starting currents of A.C. motors, in general, require oversize switches.

12. Oversize fuses or circuit-breakers necessary for large starting currents do not furnish proper protection to the motor against overloads while running.

13. The no load current of A.C. motors is high compared with D.C. motors.

Factors in the Choice of Electric Power—The foregoing analysis of the technical advantages and disadvantages of direct-current and alternating-current is based on the general factors of *reliability, performance, flexibility and simplicity.* These factors furnish no clue to the remaining factors to be considered in the selection of the type of electric power, namely, *weight, space* and *cost.* An exposition of these at this time would require predictions of competitive design and development, beyond the scope of this article.

Terminology of A.C. Machines Explained—In concluding this part of the article, it is desirable that a few definitions and explanations be made of certain terminology employed in describing or referring to A.C. motors and generators.

(a) *Synchronous*—This frequently used word is defined as *coincident—simultaneous—in step with the source of current.* Applied to *generators,* it means that a *synchronous* generator (or alternator), operates or revolves exactly in coincidence with the alternations or frequency of the generated current. The usual form of synchronous generator requires a continuous *direct-current* field. The alternating-current generated by the armature coils cutting the alternate north and south magnetic flux of the D.C. field, is fundamentally *synchronous* with the mechanical rotation. Increase or decrease in mechanical speed directly affects the frequency of the generated current. The term *synchronous* applied to a syn-

chronous *motor,* means that the motor fundamentally revolves exactly in coincidence with the alternation or frequency of the source of current. The usual form of a synchronous motor is practically identical with the synchronous generator. In both cases a continuous *direct-current* field is required. The rotation of the motor may be said to be electrically *geared* to the source of power and remains in this condition of true synchronism until an overload condition beyond its capacity forces it " out of step" which in turn results in complete stoppage of the motor.

(b) *Asynchronous* or *Non-Synchronous*—These terms which are the antithesis of synchronous, are rarely used. This is not due to the lack of application, but rather to the fact that the principle class of *motors* to which the terms may correctly be applied, are known by more specific terms based on their winding characteristics. Asynchronous is defined by the dictionary as *wanting in coincidence of time.* Electrically it means possessing no definite relation between armature speed and main current frequency. Applied to *motors,* it means that the rotation of the motor is not fundamentally based on the alternations or frequency of the source of current. This term is applied correctly to alternating-current *series* motors, *repulsion* motors and *induction* motors. These types of motors require no direct-current fields. In the *series* and *repulsion* motors the speed of rotation is a function of applied voltage and load. These motors may correctly operate at speeds from zero to many times the theoretical synchronous speed. The slip-ring *induction* motor operates correctly at all speeds from zero to within 2 to 8 per cent of the theoretical synchronous speed. As the induction motor approaches synchronism, its torque approaches zero. If mechanically driven at a higher speed than synchronous, it becomes an *asynchronous generator* for the reason that it will deliver current to the power lines without affecting the frequency of the power lines. This form of generator is used in special distribution systems in connection with synchronous generators for certain operating advantages.

(c) *Rotor* and *Stator*—These terms are used to name the two main units of alternating-current motors and generators. As the names imply, the *rotor* is the revolving unit connected to the power shaft, and the *stator* is the stationary unit surrounding the rotor.

These terms are used in a mechanical sense as either unit may be the armature or field, depending on the design of the machine.

(d) *Primary* and *Secondary*—The use of these terms arbitrarily distinguishes, from an electrical standpoint, the two principle units comprising an A.C. machine. The winding which is connected to the *power* lines is known as the *primary,* while the winding which functions by current *induced* from the primary, is known as the *secondary.* This corresponds to the same terminology as used in transformers. In all usual forms of A.C. motors, except the synchronous motor, the stator is the primary and the rotor is the secondary.

(e) *Armature* and *Field*—These terms are applied to *synchronous* generators and motors in a similar sense as applied to *direct-current* machines. The main current carrying A.C. winding is called the *armature,* and the direct-current exciting winding is called the *field.* However, unlike direct-current machines, the armature on all machines of appreciable size is the *stator,* and the field is the *rotor.* This is for the practical reason that this arrangement provides the fewer current carrying brushes necessary to conduct the current to and from the revolving winding via collector rings.

(f) *Kilowatts* (*Kw.*) *and kilovolt-amperes* (*KVA*). These terms are used to express the electrical power of generating machines. In *direct-current* machines, where the power-factor is always unity, the term *kilowatts* (Kw.) is invariably used. It is the true electrical power term. This is merely the product of volts and amperes divided by 1000. In *alternating-current,* consideration of the *power-factor* requires the use of the term *kilovolt-amperes.* This is for the reason that alternators are designed for definite rating of volts and amperes, the ampere ratings being based on temperature limitations. The term KVA, which is the product of volts and amperes divided by 1000, expresses the electrical limitations of the machine, but it is not necessarily the *true* power. The *true* power in alternating current is obtained by the product of the power-factor and KVA. This resultant is indicated directly by *kilowattmeters* installed in the mains.

NOTE:—This is the first of three articles on Alternating Current Motors and Generators; the second covers the operating characteristics of motors and the third deals with generators.

NOTES.

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Test Results and Service Values of Materials.
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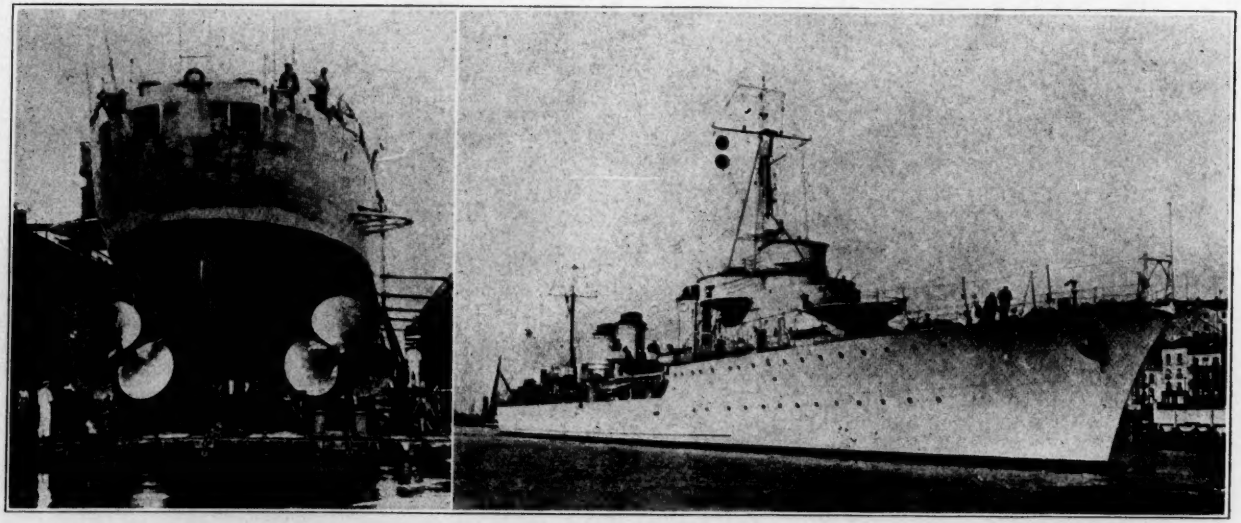
Journal of the Western Society of Engineers, Aug., 1932.



FRENCH FLOTILLA LEADER "CASSARD."

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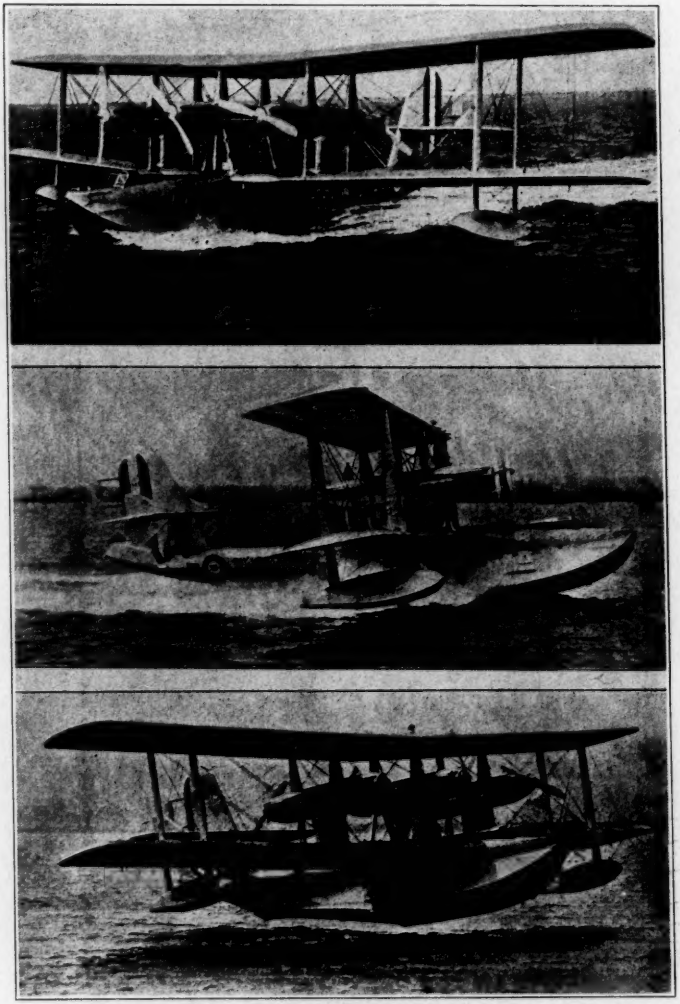

"The Aeroplane."

THE BIGGEST BRITISH FLYING-BOAT.

NEW FRENCH FLOTILLA LEADER *CASSARD*.

In the course of her official sea trials, the new French flotilla leader *Cassard*, built by Ateliers et Chantiers de Bretagne, has improved upon the speed performance set up by the *Gerfaut*, built by the same concern. The *Gerfaut* attained a mean speed of 42.787 knots and a maximum of 43.32 knots. The *Cassard* for three hours maintained a speed of 42.9 knots, and reached a maximum of 43.4 knots.

The *Cassard* belongs to a series of six similar ships ordered by the French Navy, which are 129.30 meters (424 feet) long overall, 11.84 meters (38.9 feet) broad, by 11.70 meters (38.4 feet) deep, with a Washington displacement of 2480 tons and a normal displacement of 2634 tons. The propelling machinery of the *Cassard* consists of two sets of Rateau-Chantiers de Bretagne turbines, designed to develop an aggregate power of 70,000 S.H.P., each driving a propeller shaft through single-reduction gearing. The machinery has been constructed by Ateliers et Chantiers de Bretagne in cooperation with Société Générale de Constructions Mécaniques, who were responsible for the construction of the turbines. The four Penhoët-type water-tube boilers, burning oil on the Penhoët system, have been constructed by Ateliers et Chantiers de la Seine Maritime (Worms et Cie.).

It is interesting to note that improvements have been incorporated in the new French flotilla leaders, the *Milan* and *Epervier*, now nearing completion. These vessels are fitted with high-pressure water-tube boilers supplying superheated steam to the turbines and so it is probable that a higher degree of economy will be realized.—"Shipbuilding and Shipping Record," Sept. 15, 1932.

THE BIGGEST BRITISH FLYING-BOAT.

On July 11 the Short six-engined flying-boat, officially designated the R.6/28, made her first public flight. This boat is just about twice as big as anything yet built in this country.

It has been built as a coastal reconnaissance boat, long-range bomber and torpedo-carrier to the order of the Air Ministry. It is on their Part Publication List, which means that the constructors have to preserve a proper reticence about speeds, climb, range and load-carrying capacity.

The most important publishable facts are given hereafter:—

The underwater portion of the hull is of stainless steel.

A crew of ten is carried.

There are four gun-rings, one of which is in the extreme tail behind the rudder.

There are six moderately-supercharged Rolls-Royce Buzzard III M.S. engines placed tandem-wise in three nacelles, or power-eggs. Each engine develops 930 horsepower at 2300 revolutions per minute at sea-level, so her maximum total is 5580 horsepower. The normal power is 820 horsepower at 2000 revolutions per minute at sea-level. The reduction gearing is .477 to 1.

The interesting thing about these engines is that they are steam-cooled. The radiators are tucked away under the nacelles and around the lower struts. The resistance has thus been kept remarkably low, and the equivalent frontal area for each complete unit developing about 1900 horsepower is about 8 square feet.

The maximum all-up weight of the boat is 70,000 pounds, or 56 pounds over 31 tons.

The span of both upper and lower wings is 120 feet, the overall length is 89 feet 6 inches and the height 30 feet 4 inches. And that is all one is allowed to say.—"The Aeroplane," July 13, 1932.

TRANSVERSE SHIP SECTIONS.

The writer had occasion recently to invite tentative estimates for propelling machinery for a proposed motor vessel with engines aft. An outline general arrangement was supplied, together with a statement of the principal dimensions. The lines were not available as the design was not sufficiently far advanced.

Some difficulty was experienced by sub-contractors in approximating to the transverse sections of the vessel; and this article has been prepared with a view to giving sufficient guidance to enable even a non-technical person to arrive quite rapidly at approximate sections with an adequate degree of accuracy for the purpose in question. Incidentally, the method outlined might possibly be of interest to professional naval architects, because in many drawing offices the procedure employed in roughing out approximate sections is often unnecessarily laborious.

Assume that we are given the water-line length (L), moulded breadth (B), moulded draught (D) and block coefficient (c), and that the midship section is also fixed (Figure 1).

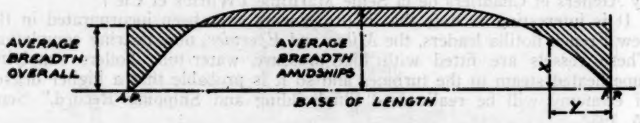

FIG. 2.—RECTANGLE OF AVERAGE BREADTH OVERALL AND CURVE OF ACTUAL AVERAGE BREADTH TO A BASE OF LENGTH.

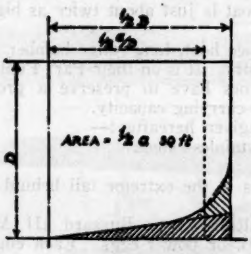

FIG. 1.—HALF MIDSHIP SECTION.

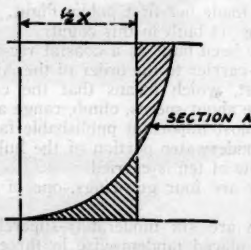

FIG. 3.—TYPICAL APPROXIMATE HALF SECTION.

Calculate the area of the midship section (a square feet) by deducting the two shaded triangles from the area D × ½B.

The "average breadth" of the midship section is therefore a/D feet. Note that a rectangle of width a/D will therefore have the same area as the midship section.

The ratio of the average breadth to the full breadth is the midship section coefficient.

Divide the block coefficient by the midship section coefficient, and multiply by the average breadth amidships. This gives the average breadth of all the

sections throughout the ship. Thus we can draw the rectangle shown in Figure 2 and the curve giving the *actual* average breadth at each section must be such that the shaded areas within and without the rectangle are exactly equal. Note that the ratio of the average breadth overall to the average breadth amidships is equal to the prismatic coefficient.

From this curve the average breadth x at any distance y from the A.P. can be measured, and the approximate section drawn in as shown in Figure 3, so that the section area is equal to the rectangle of width $\frac{1}{2}x$.

With a little practice this method enables anyone to arrive at approximate sections in a few minutes.—"Shipbuilding and Shipping Record," Sept. 8, 1932.

A NEW DEAL IN ENGINES.

By C. L. LAWRANCE.

Commencing with the development of the automobile, and later in the development of the aircraft engine, practically without exception, over a period of more than 40 years, the engines used have been of the spark-ignition, four-cycle, carbureter type originally known as the Otto cycle. This engine type has persisted with very little change in principle but with great improvement in detail.

It looks now as if in both aircraft and automobiles we are rapidly approaching the time when a great many modifications in this engine will take place, or where possibly an entirely different form of engine, such as the compression-ignition type in both two-cycle and four-cycle models, may eventually supersede it.

There are two schools of thought: the first, which seems most prevalent in this country, is that the four-cycle, spark-ignition, carbureter engine has reached such a degree of perfection that it will be very hard to induce the public to accept anything else and that it can be further improved and modified so as to maintain its position of leadership.

The use of superchargers is becoming increasingly popular in aircraft engines and even is being adopted in some high-class motor cars; and the idea of abandoning the carbureter in favor of solid-fuel injection either into the inlet manifold or directly into the cylinder probably will come into vogue, with greatly improved distribution and horsepower.

Experiments at Langley Field with greatly overlapping inlet and exhaust valves, in conjunction with supercharging and solid injection, show that great improvement in power output can be made in both aircraft and automotive engines. In addition, the development and use of so-called safety fuels will lead to much less fire risk than obtains at present.

Another school of thought, which is more prevalent in Europe than here, is that the compression-ignition engine is the coming powerplant for all automotive vehicles. This is manifested in this country most strongly in experiments on motor-trucks and motorcoaches in which the Diesel, or compression-ignition, engine is being substituted for the older type.

The compression-ignition engine lends itself to two-cycle operation so well, as has been demonstrated in large ship and stationary powerplants, that many believe that the engine of the future will be of this type. That the weight of such an engine need not be excessive has been demonstrated to a certain extent by Junkers, whose double-piston, two-cycle engine does not weigh per horsepower too much more than the average aircraft engine.

What the future will bring is hard to predict, but it is certain that the present type of aircraft and automobile engine is passing in favor of new and improved types which will lower the cost of commercial transportation and increase its speed and efficiency.—" S. A. E. Journal," September, 1932.

THE METALS OF OIL-ENGINE CONSTRUCTION.

General economic conditions are not the sole cause of the recent notable swing toward broader application of Diesel power. The Diesel came into existence as a means under certain conditions, of increasing the coverage of the power fuel dollar. Today its application is spreading largely because of operating dependability and low maintenance.

Addressed entirely to present and potential users, rather than to designers, this article shows how Diesel manufacturers are reaching into every engineering field for metals particularly adapted to the severe service conditions under which their engines must operate.

A vast store of metallurgical and practical knowledge of materials has been gained in the design and operation of gas engines. This has been particularly true since the tremendous development of the automotive engine. Because Diesel metallurgical problems are similar in many respects to those encountered with other types of prime movers, much of the knowledge gained in other fields could be applied to Diesel construction after necessary allowances were made for greater forces and higher temperatures in the combustion chamber. Consequently, the materials, and particularly the metals, of oil engine construction are in a more advanced stage of development than is generally understood.

Applications of various materials to specific Diesel engine uses can best be illustrated by considering the specific parts.

Diesel frames are cast of iron, steel or aluminum, or are fabricated of welded steel. Most have been of cast iron, because it is best known, easy to machine and comparatively low in cost. But harder service for the Diesel has necessitated stronger and yet lighter frames, and foundry technology has made possible castings ranging from straight-silicon to high-test alloys as required for particular services. Aluminum frames are used on engines where minimum weight is desired, such as in marine service.

Another method of reducing weight materially is by welding together rolled steel plates and sections. Light weight, low cost and speedy erection without patterns are advantages of welded construction. Its present disadvantage is higher cost in complicated sections and possibly where a number of units, justifying pattern cost, are to be built. Indeed, since rolled sections of maximum unit strength are used and stress relieving is employed, bed frames may often be made lighter by welding than by casting in aluminum, with equal strength. Earlier difficulties with welding have been overcome and conservative engineers now accept it as a dependable method of joining. European builders have thus far led in application of welding to Diesel construction, but a number of American manufacturers have begun its regular use.

In smaller units, the cylinders are usually cast with the top frame; in larger units they are usually made separately. This offers possibilities in the combination of welding and casting; the Diesel buyer, before long, may receive an engine composed partly of welded sections and partly of cast ones, or of rolled sections welded to cast parts.

Difference in materials for cylinders rests both with application and type of engine. If the cylinder is to be without a separate liner, it will be of a

cast iron, which provides the desirable maximum resistance to internal wear and a close, non-porous structure. If the cylinder is separately lined, either cast iron, welded steel or cast aluminum alloy construction may be used, the only requirement being that the part shall possess sufficient strength and have water-jacket surfaces suited to resist scale formation and speed heat conduction to the cooling fluid. If of aluminum, the water jacket must be properly protected against corrosion from impure cooling water.

Cylinder wear is an important consideration in economical Diesel operation. Liners were evolved for reasons of economy. In addition to the design considerations which control lubrication, ring pressure and other elements, if the liner is slow to conduct heat away from the cylinder it undergoes considerable wear by oxidation. This necessitates that it be as thin as possible.

Liners are either of the dry type (exterior surfaces protected by inclosing cylinder wall) or of the wet type (exterior surfaces exposed directly to cooling fluid). Dry liners are confined principally to small sizes because of the thicker section of the two walls and the difficulty of providing the close fit necessary for good heat conductivity at all temperatures.

Most cylinder liners are of cast iron, though some are now being made of steel and hardened Nitraloy. Cast-iron liners are dense in structure, low in silicon to resist "growth" and often include nickel and chromium to better their characteristics. Greater denseness and uniformity have recently been achieved by casting liners, centrifugally, which is a process of throwing the molten metal by centrifugal force against the walls of the mold. This results in a more compact or "packed" structure.

Another method of increasing density is by using a steel capable of being heat treated. Nickel and chromium added to cast iron will result in very hard liners. High-carbon steels have been used successfully where properly lubricated. These latter permit thin sections and consequent excellent cooling. Ni-Resist cast iron is used for liners in aluminum cylinders because its coefficient of expansion is about the same as that of aluminum, it is resistant to attack from heat and normal gases and takes on a "glaze" in service which protects it from wear. Nitraloy is used on smaller engines with good results.

Cylinder heads are usually cast iron, although steel is sometimes used on larger units and aluminum on smaller. The principal requirement is that the head be sufficiently strong to resist combustion pressures and of a material that will not be corroded by the air and gases of operation. Head sections of single-acting engines are usually complicated, because of the manifold passages and water cooling chambers. Consequently they are usually cast. Cast iron heads generally provide their own valve seats. Aluminum heads require insertion of aluminum-bronze, Ni-Resist or other special seats.

Pistons offer a difficult assignment for any material. They must be light in weight, high in strength, have good bearing characteristics and an expansion rate comparable to that of the liner, good heat conductivity, and resistance to attack by air and gases. No single material can meet all these requirements, hence the one selected is a compromise. Cast iron is usually used because its coefficient of expansion is the same as that of a cast iron liner. Its bearing characteristics are good and it is comparatively inexpensive. It has been found that straight-silicon irons oxidize readily in contact with hot oxidizing gases, and if there is a tendency for a local hot spot in the piston center they may fail rapidly from growth and fatigue. Nickel or nickel-chromium iron delays this type of failure.

Aluminum is being used extensively for pistons in smaller high-speed units and may eventually replace iron on larger sizes where water cooling would otherwise be necessary. Aluminum has a rather high coefficient of expan-

sion. In smaller engines, builders control this by using inserted Invar rings and by designing to avoid the tendency to slap when cold. Expansion is sometimes taken care of by using a split skirt. While not so likely to seize as iron, aluminum must be well lubricated.

Piston rods in a double-acting Diesel must be kept relatively small in order to interfere as little as possible with the capacity of the lower combustion chamber. This means that working stresses must be correspondingly high. Rods also may act as carriers for the piston cooling medium. If water is used for cooling, the matter of corrosion must be considered. Because of high working stresses and wear from packing, the natural tendency is to make the rods of alloy steel, suitably heat-treated to develop strength. However, these rods do not seem to stand up as well as those of softer steels because they apparently lack the inherent toughness which is essential in resisting progressive failure. Either straight carbon or some soft alloy of the nickel-manganese type seems to work best.

Piston rings are usually straight-silicon cast iron regardless of the type of liner used. Excellent results have been obtained with nickel iron. Steel of approximately spring temper may be used in smaller sizes.

Chief requirements of a piston pin are that it be light, have sufficient strength to carry the load and proper hardness to resist wear. With adequate lubrication and suitable fit, a piston pin should resist wear indefinitely. Pins may be made of hardened tool steel, although they are usually carburized and hardened steel. Nitrided and chromium-plated pins have also been used with considerable success. Bearing bronzes of the composition type are most generally used for bushings.

Crankshafts are usually single-piece forgings which have the desirable characteristics of grain flow following the contour of the section and being continuous throughout its length. In order to obtain the necessary rigidity in the crankshaft, the section is large, hence working stresses are low, and consequently the question of strength is not serious. Most crankshafts are of straight carbon steel, although on some higher speed units a harder material may be desirable. In such cases the bearings may be hardened by case carburizing or nitriding.

Most connecting rods are of carbon steel, suitably treated, although on some smaller, high-speed units, alloy steels may be required. Aluminum may be used, but often the expense is not warranted.

Bearing conditions in a small, high-speed Diesel may be quite severe, but in the usual unit they are normal. As a rule, crankshaft bearings, both main and connecting rod, are of a thin layer of white metal backed with a strong steel shell. Tin-base alloys, because of their greater strength, are preferred. The serious disadvantage in the use of most of the white metals is their low temperature operating range. When temperatures are too high for white metals, leaded bronzes may be used. Their operating limits, reached through a fusion of the lead, are in the neighborhood of 600 degrees F. Because of a high coefficient of expansion, these materials must either be used as a thin lining on a steel back or fitted with appreciable clearance. Seizure, if it occurs, is positive, but usually not destructive.

Leaded bronzes are used frequently on small, plain bearings and in guides where conditions of lubrication are difficult. In the presence of dirt, however, they tend to load and act as a lap. Camshafts and other accessory bearings are taken care of by any one of a number of the standard copper-tin-lead mixtures. Hardness must be adjusted to the mating steel.

Exhaust valves, perhaps the hottest spots in the engine, must be of a material which will withstand the temperatures. In large engines, cast iron does well, particularly if alloyed with chromium and nickel. In small units where conditions are more severe, better materials such as chrome-nickel-

silicon steels may be necessary. Ni-Resist, stainless steel and the straight chrome-nickel steels are also used, or a heat-resisting alloy head may be welded to a carbon steel or alloy stem to reduce cost.

Inlet valves, cooled continually by incoming air, require no particular attention. Good results are obtained with cast iron on the larger units and with straight alloy steels, such as carbon-chrome-nickel, on smaller. Clearance between the valve stem and guide must be small, consequently conditions for lubrication are not good. As a result, valve stems must be made as hard as practicable. Attempts have been made to do this through chromium plating, but results have not been overly successful. Guides are, as a rule, of medium hard cast iron or bronze. Valve seats are usually cut directly in the cylinder head material.

Because of the high pressures involved, and at times the corrosive nature of the oils handled, spray nozzles and pump parts must be of a hard, corrosion-resisting material. Stainless steels, suitably heat-treated, seem to meet requirements very well.—"Power," Sept., 1932.

USES OF NITRIDE HARDENED STEEL.

By Dr. W. H. Hatfield.

A great advance was made in the art of producing hardened steel surfaces when, in 1921, Fry discovered that a hardened surface could be prepared by a single heating in ammonia gas at 500 degree C., which is only a black heat. This result was achieved by using a steel of special composition, containing about 1.5 per cent of chromium and about 1.0 per cent of aluminium. Machlet, in 1913, had indicated that ammonia at such temperatures would give a hardened surface, but it was left to Fry to determine, after experimenting with many compositions, that an effective result, i.e., a surface of very great hardness, could only be obtained when the steel contained both chromium and aluminium.

The case-hardening treatment, as is now well known, consists in heating the articles in an atmosphere of ammonia at a temperature of 500 degrees C., for periods up to 96 hours. No further heat-treatment is necessary. The surface of the article should be clean and the flow of gas constant. In cases in which certain portions of the surface are required to be left soft, such parts may be protected from the nitriding action by tinning.

The special steels used for nitriding have the general characteristics of high-tensile alloy steels, and by varying the carbon content and the heat treatment, wide variations of the mechanical properties can be obtained, as will be seen from the following tables:—

	Grade 3.	Grade 5.	Grade 7.
Carbon	0.4–0.45	0.26–0.31	0.18–0.23
Manganese	0.5–0.6	0.5 –0.6	0.5 –0.6
Silicon	0.2–0.35	0.2 –0.35	0.2 –0.35
Chromium	1.5–1.7	1.5 –1.7	1.4 –1.6
Aluminium	1.0–1.2	1.0 –1.2	0.9 –1.1

With the normal nitriding treatment the depth of case produced is of the order of 0.025 inch. This case possesses great resistance to wear and is extremely hard, having at normal temperatures a diamond hardness number of 1000-1200. This hardness is maintained to a remarkable degree up to a temperature of about 500 degrees C., whilst, even at 650 degrees C. the diamond hardness is still 600. On subsequent cooling to normal temperatures

a hardness approximating to 1000 on the diamond scale is regained. Accompanying this extreme hardness there is very little capacity for plastic deformation. It has sometimes been found that when the full-time nitriding treatment is given, the outer film of the hard surface, up to about 0.002 inch thick, may be somewhat brittle, and in such cases it is desirable that a light grinding or deep polishing should be applied with a view to removing this brittle surface.

During the process of nitriding, a slight growth or expansion of the sample takes place. This expansion arises as a result of the changes in the nature of the surface material by nitriding. The amount of this expansion is therefore a function of the case depth and time of treatment. Experience shows that the amount of this growth under standard nitriding conditions does not differ appreciably from 0.0005 inch per side, or 0.001 inch on diameter. Experimental work under varying conditions has been carried out by Hobrock. He shows that the case proper increases the thickness by a fraction of itself, of the order of 4 to 2½ per cent, decreasing as the time of treatment increases. Thus, for a case of ½ millimeter depth, 2½ per cent would be equivalent to an increase in thickness of the surface layer of about 0.00048 inch. It has also been found that the rate of thermal expansion of the skin is slightly lower than that of the core, the difference over the range between ordinary temperatures and 200 degrees C. being of the order of 12 to 14 per cent.

As regards corrosion, specimens of nitride-hardened steel show very marked resistance, not only in running water, but even when partially immersed in sea water, whereas untreated specimens corrode readily in running water. The resistance to acid attack, however, is extremely small, and tests have shown that this steel, both in the nitrided and untreated condition, is useless for dealing even with cold dilute (0.1 per cent) solutions of hydrochloric or sulphuric acid. Nitriding appears to lower the resistance of the material to acid attack and nitrided specimens immersed in a 0.1 per cent solution of nitric acid suffered a loss of the order of 0.001 grams per square centimeter. The attack in 0.01 per cent solution of these acids was very small, and it is possible that in such very dilute solutions the steel might have a satisfactory life.

The addition of 0.5 per cent of copper sulphate or ferric sulphate to a 0.5 per cent solution of sulphuric acid, not only failed to prevent attack in the cold, but appeared slightly to accelerate it, while the addition of 2.5 per cent of ferric sulphate increased the attack most definitely.

A test carried out by a well-known oil company, wherein nitrided specimens were exposed to damp sulphur dioxide for a period of seven days, showed the material to be quite unsuitable for such conditions.

Immersion in a 5 per cent solution of caustic soda showed little attack on the nitrided steel, even at boiling point, nor was the attack in 25 per cent and 50 per cent solutions very great at 80 degrees C., but in the 50 per cent solution at boiling point the attack was 0.0015 grams per square centimeter in twenty-four hours, which is excessive for steels in which it is essential to preserve the surface layer.

Further tests carried out by the oil company referred to above showed the behavior of the nitrided material in contact with the vapor and condensed liquid of various boiling oil distillates to be extremely satisfactory, and it is concluded that nitride-hardened steel will be satisfactory for dealing with light petroleum spirit. Immersion tests in crude oil at 20 degrees and 80 degrees C. also showed the material to have excellent resistance to such conditions. In contact with certain packings, however, corrosion has been accelerated. Graphite packings appear to be the worst from this point of view, and the use of this packing on nitralloy steel should be avoided wherever possible.

	Grade 3.			Grade 5.			Grade 7.		
Oil Hardened, degrees	850 C.			900 C.			925 C.		
Tempered, degrees	550	625	725	550	625	725	550	625	725
Y.P., tons per square inch	69	57	51	64	51	44	41	35	32
M.S., tons per square inch	76	64	56	70	57	48	44	38	35
Elongation, per cent	12	21	25	15	22	30	27	33	37
Izod Impact, foot-pounds	32	42	54	34	50	62	76	86	101

The considerable industrial development of this process and its application for a number of years to a wide range of parts, has greatly augmented our knowledge of the properties of the nitrided case. It is now generally possible, as a result of the experience gained from the actual behavior of nitrided steel parts in service, to foreshadow the suitability of this relatively new material for any proposed application.

Bearing in mind the foregoing properties, it will be seen that nitride-hardened steel is most likely to be satisfactory where resistance to sliding abrasion is the deciding factor. The comparative thinness of the case, and the low ductility accompanying its intense hardness do not permit of high localized pressures or impact. It may be taken that the mechanical characteristics can be chosen to compare with those obtainable with ordinary alloy steels, the nitrided steel having the advantage of the high resistance to abrasion, so that it can be safely recommended for articles which have to withstand steady or alternating stresses, as well as abrasive action, provided, of course, that the stresses imposed are not too severe for the normal strength of the material. Illustrative of such applications, where the material has already proved advantageous, are motor car crank shafts, pump spindles, gear wheels, pinions and impeller pump shafts.

It has been mentioned that the nitrided case will not withstand severe local pressure or impact, and this limits its application in what would otherwise be eminently suitable conditions. It has been claimed that the surface layers of fully nitrided articles can withstand pressures up to 65,000 pounds to 100,000 pounds per square inch (depending on the hardness of the core material). Actual tests on bearing pressures have indicated that pressures up to 3 tons per square inch may be withstood. It is not recommended, however, that the bearing pressure should exceed 2 tons per square inch. Examples of applications in which nitride-hardened steel is precluded on account of severe local pressure are knife edges carrying heavy loads, cutting tools, heavy-duty ball races, and rolls for heavy-duty cold-rolling mills.

As would be expected from the low ductility of the case, the effect of localized impact is more disastrous than the effect of local steady pressures. For this reason, the barrelling process cannot be used for the manufacture of nitrided steel balls, and many applications to such parts as tappets, dies, and beaters have been unsuccessful. It should be mentioned, however, that several successful reports have been received concerning tappets and dies for certain purposes, and in such cases it is difficult to say beforehand whether nitrided steel will be successful or otherwise.

While the amount of distortion in nitride-hardened steel parts, as a result of the treatment, is in general small, considerable distortion may take place if the section of the treated piece is not symmetrical. Serious distortions in crank shafts and spindles have occurred when a single key-way only has been provided. Where possible, therefore, in such cases, two key-ways at opposite ends of a diameter should be made.

The growth of the nitrided skin has a pronounced effect on bars with central holes, especially when the wall is thin. The hole may, in such cases, expand by as much as 0.005 inch, and allowance in machining or fabrication should be made to balance this expansion.

The difference in thermal expansion between case and core is usually not serious, the difference being in the direction of relieving the stresses left in the material when cooled from the nitriding temperature. Where severe overheating occurs in service, however, the effect may be to cause cracking of the case, and subsequent failure.

One difficulty presents itself which arises from the high aluminium content of the steel used for nitriding, as such steel usually contains inclusions of alumina. These inclusions for most purposes have no detrimental effects, but there are a few applications in which their presence is undesirable. In these applications, it is essential to have a perfect mirror polish on the surface of the steel, and freedom from any scratching effects when two such surfaces are rubbed together. The use of nitride-hardened steel for precision gages of the Johanssen type has been criticized from this point of view. The presence of the inclusions also has the effect of impairing somewhat the "wringing" properties of the gages.

Nitride-hardened steel does not work satisfactorily in some bearing metals, those containing lead proving unsuitable in certain cases. Bearings of the duralumin type, phosphor bronze, and air-hardening nickel-chrome steel, however, have been found satisfactory.

One aspect of the use of such steel, where resistance to abrasion is the important factor, is worth mentioning. Accompanying the reduced wear on the parts made of nitrided steel, there is in most cases also reduced wear on the part working in contact with it. This is a very desirable feature, and renders the use of the material doubly advantageous. In this connection, brake drums and shoes made from nitrided steel have given excellent results.

It has been said that the nitrided case is likely to flake, and this is true under certain conditions. This trouble may be due to severe local pressure or localized impact, as previously mentioned, or it may be caused by severe and particularly rapid heating and cooling, since the skin expands more slowly that the interior. Flaking may further be due to severe bending or other stresses which are of a local nature in excess of the elastic range of the case. These effects naturally limit the application of the process in certain directions. It will, however, sometimes be found that the use of the harder grade of core material will reduce the effects of local stressing. One interesting case where lack of success, due to tensile stress at the surface, prevented further application, may be described. This arose in the application of nitride-hardened steel to tinning rolls. For these it is necessary for the tin to adhere to the rolls. While it did so in the cold and when moderately heated, yet at the tinning temperature in contact with palm oil the tin came away and the work was not successful.

Where there is very heavy wear, as, for instance, in grinding plates, sand-blast nozzles, &c., the thickness of the case is insufficient. For such purposes thicker cases are necessary, and it will be realized that with nitrided steel, when the case has been worn through at one spot the subsequent life is short.

Die blocks for automobile parts, particularly for connecting-rods, have given remarkable performances, while die blocks for spanners have given record production, two or three times greater than the previous best results. Dies for such substances as resinous molding compounds have proved to be a great advance, while dies for brick presses have given as much as five times the life of the previous best.

Screw and plug gages have been a great success. Pinions for small reduction gears and other types of pinions and gears have proved very effective.

In connection with the manufacture of cement, nitride-hardened steel has found a number of permanent uses, as, for instance, in screw conveyor journals and caps, packing machine levers and cover plates, grooved rollers, corrugating rollers, pump air rings, striking plates, links and pins, slurry nozzles, &c. The intensely hard surface made certain the success which has been achieved in the case of carbon dioxide compressors, pneumatic cylinder parts, &c. Numerous pins required to resist wear have been ideal applications; while for gudgeon pins of various types, the process has been largely adopted. Many types of spindles may be made from this material with much advantage.

With regard to automobile engineering, in some of the largest production units in Europe and America many parts are standardized in nitride-hardened steel.

Although essentially and mainly success in the foregoing cases is achieved as a result of the exceptionally hard case, yet in certain instances the result is assisted by the increased resistance to corrosion—Supplement to "The Engineer," Aug. 26, 1932.

TEST RESULTS AND SERVICE VALUES OF MATERIALS.

By H. F. Moore.*

This paper is presented from the viewpoint of one who, for some 27 years, has worked in two materials testing laboratories connected with two engineering schools. In both those laboratories there was a considerable contact with manufacturers and users of materials, and, in the aggregate, quite a large number of commercial tests and investigations were conducted for outside firms. It is hoped that this experience has given the writer some knowledge of, or at least sympathy with, the manufacturer and user of materials of construction, and, on the other hand, with the attitude of the research laboratory. The writer's experience has been mainly with metals, and the examples chosen for illustration are mainly from this part of the field. He wishes to assure concrete men, timber experts, textile and rubber experts, and brick and tile men that any apparent slighting of the materials in which they are interested is due to lack of intimate information on his part, not to lack of appreciation of the importance of those materials.

The pure scientist of the twentieth century is primarily, in fact almost wholly, interested in ascertaining facts—at least such is the claim frequently made by pure scientists and for pure physicists. On the other hand, the philosopher, following the path marked out by the great minds of Athens, is primarily interested in determining values—the things which are worth while, desirable and beneficial. Facts and values may be regarded as two phases of the substance of knowledge. Now the engineer, and emphatically the materials engineer, can not ignore either of these phases. He must have knowledge of facts, knowledge never exact but of the highest degree of precision available, and he must know the significance of the facts.

We may illustrate this two-fold interest by consideration of the process of deciding whether a given lot of steel may be used in the construction of automobile crank shafts. The engineer who is to make the decision must know that specimens of the steel have been taken in a certain specified way. He must know that chemical analyses and physical tests have been made in a certain definite manner. He must know the quantitative results of these

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tests, and he should have some idea of the degree of precision with which these results have been obtained. So much for his facts. He must then decide whether these results indicate that the steel is sufficiently strong, ductile, and free from impurities to be safely used in making automobile crank shafts. Usually he will be guided in this judgment of values by definite specifications which have been drawn up by groups of people including steel manufacturers and automobile builders, but the decision as to what particular group of specifications to use—or whether to abandon all specifications and trust to the manufacturer's reputation—must be made by him.

WHAT DO TESTS SIGNIFY?

As a matter of fact the determination of the significance of test results is a much less definite problem than the problem of making tests. If we could wait until materials having a wide range of test results had been tried out by years of service we could then judge of their value by the direct and empirical results of experience, but we can not wait that long, so we devise tests which can be carried out in a short time and which we judge (or hope) will show the presence, in sufficient quantitity of those properties which will make our material serviceable. All ordinary laboratory tests are by their very nature accelerated tests—tests which greatly shorten the slow process of experience. If such short-circuiting of experience were not necessary the industrial testing laboratory would also be unnecessary.

Now in the testing world the fact is that the interpretation of an accelerated test is always a matter of more or less uncertainty. On the other hand, in our modern world there has grown up what is almost a popular superstitious reverence for test results obtained by the use of apparatus on which results may be repeated over and over when that apparatus is handled by skillful laboratory technicians. May I illustrate this by an incident which occurred during the early years of the War.

In 1917 the use of autogenous welding was increasing by leaps and bounds, and the question of its use in ship building was of great importance. But there were welds and welds. How could the difference between good ones and bad ones be determined? One enthusiast at a committee meeting proposed that welds be tested by cutting a specimen strip from a welded plate and that this specimen strip be violently twisted back and forth through a given angle and the number of twists before destruction be counted as a quantitative figure of merit for the weld. He argued that this would be a quick, simple test which could be carried out with precision and which would give clear-cut quantitative results. When asked if this test reproduced the destructive actions in ship plates he replied that ship plates were subjected to complex and unknown stresses and in the proposed test the specimen would be also subjected to complex and unknown stresses—a statement which was quite true.

A representative of a large government bureau who was present then satirically suggested a still simpler test. His test was to consist of taking a number of specimens from a welded plate, drawing two chalk lines 15 feet apart on a horizontal floor, and then, standing with the toe of the right foot on one line, pitching the specimens one by one at the other line. The distance that each specimen fell from the chalk line was to be measured and the average measure was to be taken as the measure of the quality of the specimens. This plan also subjected the specimens to unknown and complex stresses and gave quantitative results determined with a good degree of accuracy.

Neither test plan was adopted.

I have noted this rather ridiculously extreme case of a precise but meaningless test, but there are tests in actual use in which the precision with which the test results can be obtained is far higher than the degree of certainty as to the significance of the tests as indications of service value. It seems in order to quote three examples of such tests.

MANY TESTS ARE INCOMPLETE.

First may be mentioned the various tests for ductility, measurements of elongation, reduction of area and of various bend tests. These tests show the behavior of a specimen of metal when subjected to one extremely violent loading, resulting in complete fracture or extreme distortion. In selecting metal to be used in drawing and rolling operations this ability to withstand extreme punishment by stretching or bending would seem to be some indication of the service value of the metal, but even here the question of the effect of speed of deformation is of great importance for some metals, notably for zinc.

When we consider ductility as an index of value of steel to be used in a structure we must face the fact that our specifications demand an enormously greater degree of ductility than the metal would ever have to withstand in a structure, and that it seems difficult to conceive a case where a structure could withstand more than 4 or 5 per cent deformation along any dimension without failing to function.

Still more difficult is the question of the significance of test results for ductility when we come to machine and structural parts subjected to repeated loading. Under repeated loading service one of the requirements of service is that at localized spots of high stress the metal shall be able to readjust its stress distribution without starting a crack. The ordinary tests for ductility do not give very much information of the ability of a metal to undergo such localized plastic readjustment cracklessly. The higher grades of alloy steel when heat-treated show good elongation and reduction of area but under repeated loading involving plastic action develop minute spreading cracks rather readily. There is need today for tests to determine the service value of the material under repeated high localized stress. Such localized stresses occur at the edge of rivet holes, the root of screw threads, at the corners of keyways, etc.

TESTS NEEDED FOR CORROSION AND IMPACT.

Perhaps no service value of materials, especially metals, is so hard to determine by test as is resistance to corrosion. There are many kinds and degrees of corrosion and short-time high resistance to severe corroding agents can be determined quickly and accurately, but this determination is of no value in determining long-time resistance to a mild corroding agent of another kind. At present all accelerated corrosion tests are regarded as of very doubtful realiability in determining service values.

"Impact" has always been a mystery word in engineering mechanics, and failures whose cause could not be determined are frequently laid to the effects of "impact." At just what speed of application of load the mechanical resistance of material begins to change is a problem which has a different answer for every different kind of material, and has, at the present time, a definite answer for none of our ordinary materials, with the possible exception of timber. It appears simple enough to test resistance to impact by subjecting a specimen of a material to a violent impact from a swinging pendulum or a falling weight, but is there any reason to expect that the

effect of comparatively slow application of the load from a moving train to the members of the truss of a bridge can be judged by the far more rapid application of energy from a Charpy or Izod pendulum to a small specimen.

Again, in order to make a small specimen of ductile metal break in bending, it is deeply notched, and it may well be that the test indicates more the effect of the notching of the specimen than it does the effect of the rapid application of the load. While the making of an impact test on a notched specimen is a very simple matter, the significance of this impact test is a matter of much complication and uncertainty. The test is so easy to make, so inexpensive, and so rapid and it gives such definite test results that every testing engineer yearns to use it. Moreover, there seems to be reason in the general proposition that a metal which will withstand a large amount of energy applied by a blow must thereby be credited with some kind of superiority over a metal which will withstand only a small amount of energy.

Still further there is slowly accumulating a body of service experience to the effect that in certain lines of work there have been no failures in parts made of metals which have shown high values under the notched-bar impact test. In these fields all unexplained failures have been reported for metals whose impact test values were low. It is to be hoped that these data of experience can be gathered together and summarized, perhaps by the A. S. T. M. Impact Committee. Perhaps still more important, it has been found that this notched-bar impact test seems to be a particularly significant indicator of certain heat treatments which are generally recognized as giving poor service results.

I mention this notched-bar impact test as an excellent example of a test of uncertain significance developed rapidly because it was a convenient and simple test to make—a test whose significance is gradually becoming discovered as the test is used. This to a Greek philosopher would seem to be a most deplorable method of securing knowledge. However, the engineer is less squeamish as to methods of acquiring knowledge and follows Galileo, who obtained his ideas of strength of materials from watching the workmen in the Venetian shipyards and the accidental failures in the marble cutting yards.

BENDING TESTS FOR BRITTLE MATERIALS.

Perhaps we may now consider several typical laboratory test results, their accuracy, their uniformity, and their significance. First let us consider tensile strength, the steady pull per square inch of cross-section which a material will stand before fracture. The accuracy of this test value is usually very high for ductile metals and with uniform metals it can be repeatedly obtained within a very close range. In the Standards of the A. S. T. M. there is recognized a rather crude method of calibrating testing machines by determining tensile strength of specimens of ductile metal tested on a standard machine and of other specimens cut from the same bar tested on the machine to be calibrated. Individual test results for each machine are discarded if they do not come within 1 per cent of the average results for a set of specimens. For some three years the speaker has given this calibration test as a class exercise in his advanced laboratory practice. During that time some 25 or 30 sets of specimens have been run and only 2 or 3 individual results have had to be discarded.

In the case of brittle materials the determination of the tensile strength is far less exact. The yield and plastic flow of ductile materials tends to make the stress on the cross-section of a specimen uniform, so that the

assumption of uniform stress, which is made in computing tensile stress, is very closely correct. In brittle materials, on the other hand, there can be very little of this plastic readjustment and it has been found extraordinarily difficult to secure axial tension and uniform stress distribution for such materials. This is especially true for short specimens. Even if the specimen is gripped evenly all around and loaded axially there is a considerable unknown variation of stress from axis to surface of specimen. For this reason tensile tests are not very much used for brittle materials, and when they are made they are so sensitive to variations in gripping that the results are apt to show a pretty wide scatter. In general, cross-bending tests give better results for brittle materials than do tensile.

VALUE OF TENSILE TESTS.

For ductile metals the tensile strength does not have any very direct significance. With the exception of chains and bolts, it can rarely be developed without such distortion that the piece fails to function. Indirectly, however, it seems to have considerable significance. For any given class of metals the elastic strength and the tensile strength seem to have a rough correlation and the tensile strength also seems correlated with the limiting stress for endurance under repeated loading. Both these correlations are rough but the second, at least, is serviceable.

If a brittle material were used in tension the tensile strength would be a directly significant value. However, owing to the tendency of brittle materials to snap off without warning, they are rarely used in direct tension. In the case of wood it is so difficult to devise a test piece which will fail by tension rather than by shearing along the grain that the tensile test is rarely used. In rope, wire rope and belting, the tensile strength is of very direct and evident value.

NEED TO UNDERSTAND ELASTIC ACTION.

The ultimate compressive strength of ductile materials is impossible to determine. Short compression test specimens flow and expand laterally so that ultimate strength cannot be determined. Ultimate strength is sometimes reported by an arbitrary measurement of deformation. The writer does not believe this gives any real ultimate strength, and doubts whether such a test is of much value. For brittle materials compressive strength is a fairly definite test result; though it probably represents destruction by shearing action, or by lateral strain, rather than by direct compression. It would seem to be a good indication of service strength for brittle materials under compressive loads.

The cross-breaking strength, or modulus of rupture, can be obtained only for rather brittle materials. It is correlated with the tensile strength, but owing to the fact that even the most brittle materials flow somewhat before fracture, its value is always higher, and frequently much higher, than that determined by a tensile test. This difference may be partly due to the extreme difficulty of getting a tensile test of brittle material in which the stress is uniformly distributed over the cross-section of the test specimen, and the difference is doubtless partly due to the fact that even the metals which are classified as brittle exhibit some plastic action, stress ceases to be proportional to strain, and the actual developed stress falls below the computed stress.

Unfortunately, the terms elastic limit, proportional limit, and yield point are used quite loosely to denote the limit of pure elastic action in a metal. A vast amount of discussion upon this limit of elastic action is on record. Suffice it to say that for most metals there is a region below which the action

of the metal is almost purely elastic and above which distinct plastic action predominates. The values located by various tests for elastic limit, proportional limit, and yield point, when carefully analyzed, seem to be arbitrarily determined values lying within this range. For ductile metals, and especially for structural steel, this range of stress between elastic and plastic action is quite narrow, whereas for many non-ferrous metals and cold-worked metals it is much wider.

The writer believes that it should be frankly recognized that any test for determining a limit of elastic strength should be recognized as arbitrary. He calls attention to the discussion of current and proposed methods for determining practical "yield strength" in the 1931 Proceedings of the A. S. T. M., Part I, report of Committee E1 on Methods of Testing, and to the proposed methods of test submitted by the section on Elastic Strength.

The service significance of the elastic strength is certainly very great. This is especially true of structural members under compression where any very great amount of plastic action may lead to a disastrous collapse of a whole structure. The elastic strength is also of importance in machine parts, many of which cease to function if any serious amount of distortion takes place. In brittle materials the location of elastic strength becomes almost purely arbitrary, and it is doubtful whether any such limit for any brittle material is of very great practical significance.

SIGNIFICANCE IS MORE IMPORTANT THAN ACCURACY

In this connection the speaker wishes to question the service significance of "elastic limits" determined by methods of extreme precision. In the first place it is extremely difficult to reproduce the results. Slight variations of testing methods and apparatus change the values of such delicately determined limits widely. In the second place a slight inelastic action can usually be determined at almost any stress, providing the methods used are sufficiently delicate. In the third place the writer knows of no evidence that these extremely small amounts of plastic action cause any appreciable structural damage to a member so long as they are not repeated thousands of times. The writer believes that some of the arbitrary testing methods which determine the stress at which evidence of a certain definite amount of plastic flow appears give more significant values of elastic strength than do extremely delicate methods.

The questions of ductility and of impact tests have been briefly mentioned in the foregoing paragraphs. Suffice it to note here that the question of the significance of these tests for service results is probably shrouded in much more uncertainty than is the question of accuracy of tests.

FATIGUE TEST IS IMPORTANT.

In recent years two new limits of strength have been reported from many testing laboratories. The first of these is the endurance limit or fatigue limit, a limiting stress below which a material will stand an indefinitely large number of cycles of stress without failure. This limit can be determined with a fair degree of precision, but its determination at present involves a series of tests on six or more specimens, and requires from a week to a month, on account of the long-time tests which are necessary to determine it. So far machines and methods of testing for this limit have not been standardized though a type of machine which has been in use for the past 75 years is the one most commonly used.

The significance of this limiting stress is a matter of considerable debate between testing engineers, machine designers, and steel makers. It would

seem at first that this limit has a rather direct bearing on the ability of a material to withstand repeated stress. Perhaps the clearest case is that of metal to be used in a spring subjected to a wide range of stress, with excessive deflection prevented by stops. In this case the endurance limit would seem to be of marked direct significance. It would seem to be of very considerable significance for shafts, axles, wire ropes, railway rails, and many other machine and a few structural parts. Owing to the fact that for steel this limit seems to be fairly closely correlated with the tensile strength, the fatigue test has not as yet made its appearance in specifications to any great extent, but resistance to repeated stress is judged from the tensile strength. It should not be judged by any kind of elastic limit.

In this connection it seems wise to emphasize the fact that very few structures are in danger of fatigue failure. Failure by buckling when the elastic strength is exceeded is a far more imminent danger. It is in machine parts that fatigue failures are most frequent, but even in this field it should be emphasized that many machine parts are in no particular danger of failure by fatigue. Many pressure vessels have the load on them repeated only a few hundred times, which hardly brings fatigue strength into play. The case of the steamboiler where the stress may be repeated a few thousand times in the life of the boiler seems a border line case. We may say with reference to the fatigue limit as with reference to any other proposed new test, "Do not use the test unless it seems to give results closely correlated with the service required."

CREEPING WHEN HEAT LIMITS LOADS.

The second new limit, newer even than the fatigue limit, is the creep limit. At elevated temperatures for ordinary metals (and at room temperatures for lead) under a steady load there goes on a continuing stretch or compression of the metal. This is different from the plastic flow which takes place at the yield point, for such plastic flow does not continue indefinitely under steady load. Whether for any metal there exists a limit below which the creep is zero at a given temperature is as yet uncertain. However, a limiting stress for most metals and most temperatures can be found below which the amount of creep will not be disastrous in the desired life of the member. This whole subject of creep is such a recent development that we can not say how important this test will be or whether any short-time tests for this creep will become feasible. It gives, however, promise of very great significance as a test value in connection with metals which have to carry stresses at high temperatures.

This discussion could be extended indefinitely—chemical tests, microscopic examination, hardness tests, shearing and torsion tests—and a score of other topics, offer fertile fields for consideration. However, this is a paper—not a textbook—and the writer will close with a piece of rather trite advice. Let every testing engineer keep alive his interest in and his study of the technique of testing and also of the service significance of test results—"Journal of the Western Society of Engineers," Aug., 1932.

ASSOCIATION NOTES.

The Annual Meeting of the Society was held at Washington on October 4, 1932.

The following were nominated for officers for the calendar year 1933:

For President:

Rear Admiral William A. Moffett, U. S. N.

For Secretary-Treasurer:

Commander H. B. Hird, U. S. N.
Commander C. S. Gillette, U. S. N.
Commander S. S. Kennedy, U. S. N.

For Member of Council:

Captain R. C. Davis, U. S. N.
Captain H. G. Bowen, U. S. N.
Captain N. H. Wright, U. S. N.
Captain H. M. Cooley, U. S. N.
Captain C. A. Bonvillian, U. S. N.
Commander C. A. Jones, U. S. N.
Captain Henry Williams (C. C.), U. S. N.
Commander Garland Fulton (C. C.), U. S. N.
Captain John Q. Walton, U. S. C. G.
Commander W. M. Prall, U. S. C. G.
Mr. C. P. Wetherbee.
Mr. John E. Burkhardt.
Mr. F. D. Herbert.
Mr. J. H. King.

Ballots have been distributed and will be received until the close of working hours on 26 December, 1932.

It was decided to hold a banquet during 1933. Information relative to this will be given in due time.

MEMBERSHIP.

The following have joined the Society since the publication of the August, 1932, JOURNAL:

NAVAL.

Cunningham, George B., Lieut. U. S. N. R., Swann Chemical Co., 420 Lexington Avenue, New York, N. Y.

Danke, John L., Lieut., U. S. N. R., Waterman S. S. Corp., Mobile, Ala.

O'Neil, Charles Henry, Lieut. U. S. N. R., 540 Ocean Avenue, Brooklyn, N. Y.

CIVIL.

Fowler, Thomas P., Bath Iron Works Corporation, Bath, Maine.

ASSOCIATE.

Symon, Maxwell S., 324 East 19th St., New York, N. Y.

Tracy, Charles Barnard, 7 Federal Place, Mariners Harbor, Staten Island, N. Y.

Wyche, J. W., 829 11th Avenue, Prospect Park, Pa.